SPACE AND RE-ENTRY SYSTEMS UG-4354

QUARTERLY PROGRESS REPORT

DEVELOPMENT OF ADVANCED SOIL SAMPLER TECHNOLOGY

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SECTION 1

INTRODUCTION

This quarterly progress report summarizes the results of Task E, for the period from 1 January 1968 to 31 March 1968. The intent of Task E is to generate engineering prototype designs of lunar and planetary soil samplers and mechanical sample processors. These designs were to be based on breadboard mechanisms as identified by the Jet Propulsion Laboratories, Pasadena, California. This task was scheduled to follow the completion of the laboratory testing of two soil samplers, developed at Philco-Ford under a preceeding contract, and the field testing of these samplers along with nine other mechanisms furnished by JPL. It was scheduled in this manner so that the full benefit of the test experience with these soil samplers could be utilized in the completion of Task E.

The sampler and processing mechanisms designated by JPL in the work statement are listed as follows:

(E-1) Uncased Rotary Impact Drill Sampler

(E-2) Cased Rotary Impact Drill Sampler

(E-3)* Conical Abrading Sieve Cone Sampler

(E-4)* Helical Conveyor Simple Particulate Sampler

(E-5)* Backhoe Sampler

(E-6)* Soil Auger Sampler

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* Mechanisms tested in the preceeding tasks.

(E-7) Miniature Jaw Crusher

(E-8) Sample Size Sorter

Based on subsequent conversations with JPL, the following interpretations are considered to apply to this design effort.

The purpose of this task is to use the results and demonstration of feasibility from various breadboard sampler models to develop a more advanced engineering design. These designs should emphasize simplicity and reliability. All the mechanisms listed above are identified as geological samplers and any consideration of their use as a biological sampler is strictly secondary. The design should consider the use of the sampler mechanism as a means of enhancing the successful completion of "in-situ" experiments such as was demonstrated for the α-scattering experiment on Surveyor. In this case the soil sampler mechanism was used to aid in dislodging the α-scattering experiment when it failed to deploy properly.

Additional clarification was provided by JPL on the specific design approaches that should be pursued for the simple particulate sampler (E-4) and the particle size sorter (E-8). The simple particulate sampler (E-4) was not to be a linear vertically deployed device, as tested in the field, but rather a curved helical conveyor similar in configuration to the design approach presented in Figure 1, SPS 37-46, Volume IV, page 137. The particle size sorter (E-8) design also is to pick up the basic approach of such a mechanism, developed at JPL for the petrographic microscope, as described in JPL TM33-353. The jaw crusher (E-7) design has also been redirected to apply the principles of a rotary rock crusher breadboard which has been built by JPL and undergone some limited testing. Thus, although some preliminary effort was made on a jaw crusher design, the final design of this mechanism will be a rotary rock crusher.

The design procedure used in performing this task consists of the following.

(1) Define the design criteria for each mechanism. Utilization of JPL'x experience and other sources is made in defining these criteria. JPL assists in the complete definition of the criteria for each mechanism.

(2) From these criteria and past experience with various applicable breadboard components, preliminary sketches, schematics, layouts, and block diagrams are generated for design review.

(3) From the selected design approaches, a complete layout is generated and supported with the necessary analysis to provide a design in sufficient detail to allow a mechanical evaluation of the mechanism. This evaluation includes estimates of size, volume, and weight and a qualitative estimate of simplicity.

SECTION 2

DESIGN CRITERIA

At the initiation of this task, it was apparent that in order to minimize confusion in the design goals desired for each sampler or processor mechanism, that a definitive set of design criteria should be drafted. A preliminary set of the design criteria were presented to JPL early in the effort for their review. In general these were acceptable but have since been revised and expanded to conform more nearly with the desired design objectives. The criteria for mechanisms (E-4) helical conveyor simple particulate sampler, (E-7) miniature rotary rock crusher, and (E-8) sample size sorter in particular have been reworked to reflect the specific design approaches requested. These criteria are presented in the following paragraphs.

2.1 UNCASED ROTARY IMPACT DRILL, E-1

This is an uncased rotary impact sampling drill which is based on design components of both the JPL and Hughes Tool Company models. The following initial design criteria are assumed:

(1) This drill must obtain a sample from solid rock, rubble, sand, and cohesive powders. No segregation of sample as a function of depth is required.

(2) The particle size output of the sampler shall be 250μ or less in diameter.

(3) A minimum sample quantity collected per run shall be 2 to 3 grams of soil.

(4) The sampler shall have the capability of repeating a sampling run at least one or more times in several locations, i.e., mounted on a deployment structure.

(5) Continuous sample transport will be used. The operational mode shall be such as to minimize carryover of one sample to the next by clearing the drill and conveyor between sampling runs.

(6) Power required should be 50 watts or less.

(7) The hole diameter shall be as small as possible consistent with acquiring a 2 to 3 gram sample at a reasonable depth in rock; i.e., less than 5 centimeters deep.

(8) Maximum depth of penetration in material other than rock shall be 25 centimeters.

(9) This sampler will start the drilling operation without impact after being deployed to the surface. The sampler shall be capable of sensing feed rate to indicate rock. Impact mode of drilling starts when rock is sensed.

(10) The minimum drill rate in basalt shall be .05 inches per minute.

(11) The axial thrust shall be limited to 20 pounds.

2.2 CASED ROTARY IMPACT DRILL, E-2

This is a cased rotary impact sampling drill in which the casing can be either driven or stationary. The casing is rotated and driven vertically independently with respect to the drill. This design will consider components of breadboard models built by JPL. The following initial design criteria are assumed.

(1) This drill must obtain a sample from solid rock, rubble, sand, and cohesive powder at or near the planetary surface. It shall be capable of obtaining an essentially uncontaminated rock sample from solid rock under an overburden not more than 20 centimeters thick.

(2) The particle size output of the sampler shall be 250μ or less in diameter.

(3) Minimum sample quantity collected shall be 2 to 3 grams of soil. If rock is encountered, two samples will be collected. A sample of overburden and a rock sample.

(4) The sampler shall have the capability of repeating a sampling run at least one or more times in several locations.

(5) Continuous sample transport shall be used.

(6) Power required should be 50 watts or less.

(7) The hole diameter shall be as small as possible consistent with acquiring a 2 to 3 gram sample at a reasonable penetration in solid rock; i.e., less than 5 centimeters deep.

(8) Maximum depth of penetration, if no rock is encountered, shall be 25 centimeters.

(9) This sampler will start drilling without impact. It shall be capable of sensing feed rate to indicate rock. Impact mode of drilling starts when rock is sensed.

2.3 DEEP ABRADING CONE SIEVE, E-3

This design is based on the JPL deep abrading sieve cone tested in the field. The basic concepts of the small half angle cone as a selective acquisition device will be retained in this design. The following initial design criteria are assumed.

(1) The sampler is not required to obtain a sample from solid rock. It is required to obtain a sample from cohesive soils, sand, rubble, and cohesive powder. It should have limited sampling ability on softer rock material such as sandstone and vesicular pumice. Strict segregation of the sample as a function of depth is not required.

(2) The particle size output of the sampler shall be 250μ or less in diameter. The particle size distribution obtained in the field tests for this sampler met this limit with a mean grain diameter of 100μ.

(3) A minimum sample quantity collected shall be 10 grams of soil.

(4) The sampler shall have the capability of repeating a sampling run one or more times in several locations; i.e., mounted on a short boom.

(5) Continuous sample transport using a helical conveyor or batch aerosol pneumatic transport may be used.

(6) Axial thrust during sampling shall be limited to 20 pounds.

(7) A gimbal mount shall be employed to minimize the possibility of defeat due to encountering a surface obstruction (rock).

2.4 HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER, E-4

This sampler design is based on the JPL concept of a simple particulate helical conveyor sampler utilizing a hard or elastomer lined casing and helical screw. The configuration as shown in Figure 1, SPS 37-46, Volume IV, page 187, will serve as the basis for the design. The basic concepts will be retained; however, design modifications suggested by the results of the field testing will be evaluated. The following initial design criteria are assumed.

(1) This sampler will be capable of sampling only weak cohesive materials, rubble, sand, and cohesive powder. A drill cutter shall be incorporated at the tip of the sampler to enhance its capability in soft rock materials. Segregation of sample as a function of travel is not required.

(2) The particle size output of the sampler shall be 250μ or less in diameter.

(3) A minimum sample quantity collected shall be 2 to 3 grams of soil in traversing a minimum of 5 inches through the soil.

(4) The sample shall have the capability of repeating a sampling run one or more times by reversing the motor to reposition for the next cycle.

(5) Continuous sample transport using a helical conveyor shall be used.

(6) Deployment rate shall be set to be compatible with the desired feed rate of sampler.

2.5 BACKHOE SAMPLER, E-5

This is a backhoe type sampler utilizing some form of boom deployment combined with a batch type of sample transport mode, probably a gravity dump. This sampler is considered to be primarily a surface sampler. The following initial design criteria are assumed.

(1) This sampler will be capable of sampling only weak cohesive material, rubble, sand, and cohesive fine powder.

(2) The particle size output of this sampler shall be limited to 5 millimeters or less in diameter.

(3) A minimum sample quantity collected shall be 10 grams of soil per run.

(4) Batch type sample transport using a gravity dump will be considered for this sampler.

(5) The sampler shall have a deployable range up to 5 feet and be able to sample in a sector with a minimum angle of arc of 90 degrees.

(6) Both mechanically extendible telescoping booms and furlable tape closed-section booms (Ryan) shall be considered.

(7) The backhoe scoop design shall be such that it may be closed after clearing the surface without losing excessive sample. Also, the design shall be such that the scoop can be completely closed; i.e., no rocks will be in a position to cause the scoop to be partly wedged open.

2.6 SOIL AUGER SAMPLER, E-6

This sampler is based on the breadboard soil auger developed by JPL and tested in the field. The basic concept of the auger design and sample transfer mode will be retained. This is considered to be a shallow subsurface sampler. The following initial design criteria are assumed.

(1) The sampler will be capable of sampling only weak cohesive materials, rubble, sand, and cohesive fine powder.

(2) The particle size output of this sampler shall be 5 millimeter or less in diameter.

(3) A minimum sample quantity collected shall be 2 to 3 grams of soil per run.

(4) Batch type sample transport using a spin dump mode of soil transfer will be considered.

(5) The sampler shall consider both vertical deployment and boom deployment to the surface.

(6) If vertical deployment is used, the sampler will be capable of repeating the sampling run one or more times in several locations.

(7) If boom deployment is used, it shall be a simple non-extendable short boom and be capable of being deployed in a sector with a minimum included angle of arc of 90 degrees.

(8) Sample transport from the sampling head will consider a gravity dump mode for the boom mounted sampler.

2.7 MINIATURE ROTARY CRUSHER, E-7

This is a miniature crusher designed to break down larger pebbles to a size useable by geological analytical instruments such as the X-ray diffractometer, the alpha-scattering spectrometer, or the petrographic microscope. The existing JPL breadboard design of a rotary crusher will serve as the basis of this design. The following initial design criteria are assumed.

(1) This crusher shall be capable of accepting and reducing pebbles up to 5 millimeters in diameter contained in the sample delivered to this crusher.

(2) The rock crusher output shall contain particles up to a maximum of 300μ.

(3) The maximum power consumption shall not exceed an average value of 15 watts.

(4) The crusher shall be capable of reducing hard rock such as basalt, quartz, etc., without metallurgical contamination.

(5) The crusher shall include the means of preventing particles larger than 5 millimeters from being ingested.

(6) The crusher shall minimize the probability of ingesting particles larger than 300μ which possess a high magnetic permeability.

2.8 SAMPLE PARTICLE SIZE SORTER, E-8

This processor shall selectively sort by particle size a sample delivered to it by any of the samplers or the rotary rock crusher into particle size cuts suitable for use in geological analytical instruments such as the X-ray diffractometer, alpha-scattering spectrometer, or the petrographic microscope. The JPL particle size sorter developed for the petrographic microscope as described in JPL TM 33-353 shall serve as the basis for this design. The following initial design criteria are assumed.

(1) This size sorting processor shall be capable of separating the sample introduced into it into into three cuts with $d > 300\mu$, $50\mu < d < 300\mu$, and $d < 50\mu$.

(2) The sample introduced into this size sorting processor may contain particles up to 5 millimeters in diameter.

(3) This processor will be capable of repeating the size sorting operation on samples one or more times.

(4) The maximum quantity of sample processed in any given cycle shall not exceed 10 grams.

SECTION 3

DESIGN APPROACH

This section presents the design approaches for the sampling and processing mechanisms thus far developed. The basic considerations for each design are governed by the individual design criteria and functional requirements for each mechanism. The design goals are intended to produce the simplest, most compact, and lightweight mechanism that satisfy these design criteria and functional requirements. The designs are also to incorporate the results obtained from breadboard models and any testing that was accomplished with these models. In support of the design effort, JPL has made available to Philco-Ford various sketches, drawings, and reports as listed in Table I. This information has been supplemented by means of verbal comments and suggestions by JPL during the progress of the design.

In the following paragraphs each design is discussed in a sequence following the order listed in the introduction. In most cases the designs presented represent only partially completed designs.

3.1 ROTARY/IMPACT DRILL SAMPLER, E-1

Since one of the most critical parts of this sampler is the impact hammer and rotary drive mechanism, this design was considered first. The basic approach was to determine whether or not a more compact mechanization could be achieved that would deliver the same impact characteristics of the cam actuated hammer tested by Hughes Tool Company for JPL. The basic assumptions were that a helical conveyor would be used to transport soil particles away from the bit, the helical conveyor would have to pass

TABLE I

LOG OF SUPPORTING DATA RECEIVED

Identifying No.	Originating Source	Identifying Title	Date Received
10018381	JPL	Bulk Particulate Sampler, Auger Type	8/0/67
X3024700	Hughes Aircraft Co.	Soil Mechanics/Surface Sampler, Surveyor	3/6/68
6-9376306	JPL	Lunar Drill Assy - Breadboard	3/6/68
Layout	JPL	Cased Drill - Drill Feed Mechanism	3/6/68
SK9079	Ryan Aircraft Co.	Demonstration Model - 12 Ft. Retractable Boom	3/6/68
J10002468-1	JPL	Processor Assy - Petrographic Experiment, Sheet 1 of 30	3/6/68
Final Report JPL Contract 951480	Hughes Tool Co. (Phase I & II)	Development Program of a Lunar & Planetary Geosampler	3/6/68
Layout	JPL	Impact Drill - Hammer Drive Assy	3/21/68
Layout	JPL	Impact Drill - Vibratory Feed Test Extractor	3/21/68
325-491B	JPL	Petroscopic Microscope (Photo)	3/21/68
325-467B	JPL	Petroscopic Microscope Sieve Assy (Photo)	3/21/68
2 Pages of Paper	JPL	Petrographic Microscope Description	3/21/68
29770-10	Ryan Aircraft Co.	12 Pages of Furlable Boom Proposal	3/21/68
Layout 1001	Hughes Tool Co.	B-1 Drill and Transport Assy	3/21/68
Case No. 1384 IR No. 30-1384	JPL	Particulate Auger (New Technology Report)	4/1/68
Job No. 383-20701-2-3220	JPL	Auger Blank Bulk Sampler	4/1/68
Job No. 383-20701-2-3220	JPL	Tip Detail, Saw Tooth Auger	4/4/68

through the hammer drive assembly to some point above it, and that one drive motor should operate both the impact hammer and the rotary drive of the drill. If possible, it should also provide the drive for the helical conveyor. To satisfy these assumptions, a free wheeling crank system to compress the hammer spring was considered. The crank in this system is connected to the driving source through an over-running clutch so that as top dead center is passed the spring is free to accelerate the hammer carrying the connecting rod and crank with it. After impact the over-running clutch engages the drive shaft which lifts the hammer and compresses the drive spring. The velocity and stroke characteristics for this hammer mechanism were calculated using the parameters listed in Table II.

TABLE II

IMPACT HAMMER PARAMETERS

Item	Value
Crank Speed	240 rpm
Hammer Stroke	.5 inch
Hammer Weight	1.75 lbs
Spring Rate	40 lb/inch
Preload Force	40 lbs
Compressed Force	50 lbs
Impact Energy	1.5 ft-lbs

The characteristics for three types of hammers as shown in Figure 1 were compared in order to assess what differences might exist in the operation of the hammer. The type A hammer is one that has been built and tested by JPL but not necessarily with the same parameters listed in Table II. The type B hammer is the approach described earlier and the type C hammer is a cam actuated device built and tested by both JPL and Hughes Tool Company. The velocity and stroke characteristics are shown in Figure 2 for these hammer types. The most noticeable difference is the number of impact strokes delivered per revolution of the crank drive shaft. The same rotational speed is assumed for each crank drive shaft input.

The type B hammer with the over-running clutch delivers more strokes per revolution. For this configuration the ratio is 1.7 impacts to one impact for each of the two other types. It should be pointed out that the calculations are made neglecting friction in the system and making no allowance for clearances resulting in lost motion. Thus, the hammer velocity at impact is the same for all three types. In this case it is 8 fps. The

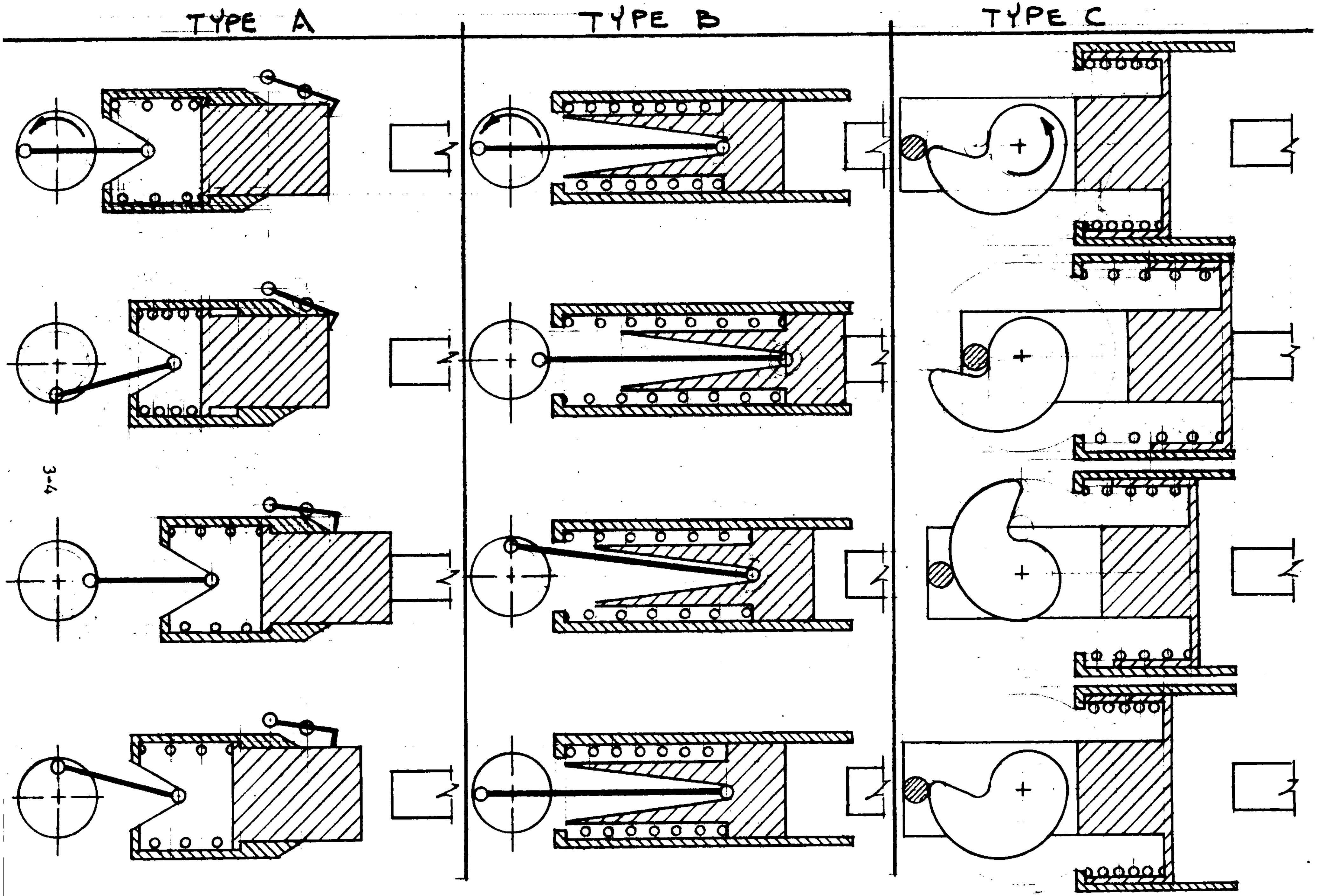


FIGURE 1 SCHEMATIC - IMPACT HAMMER TYPES

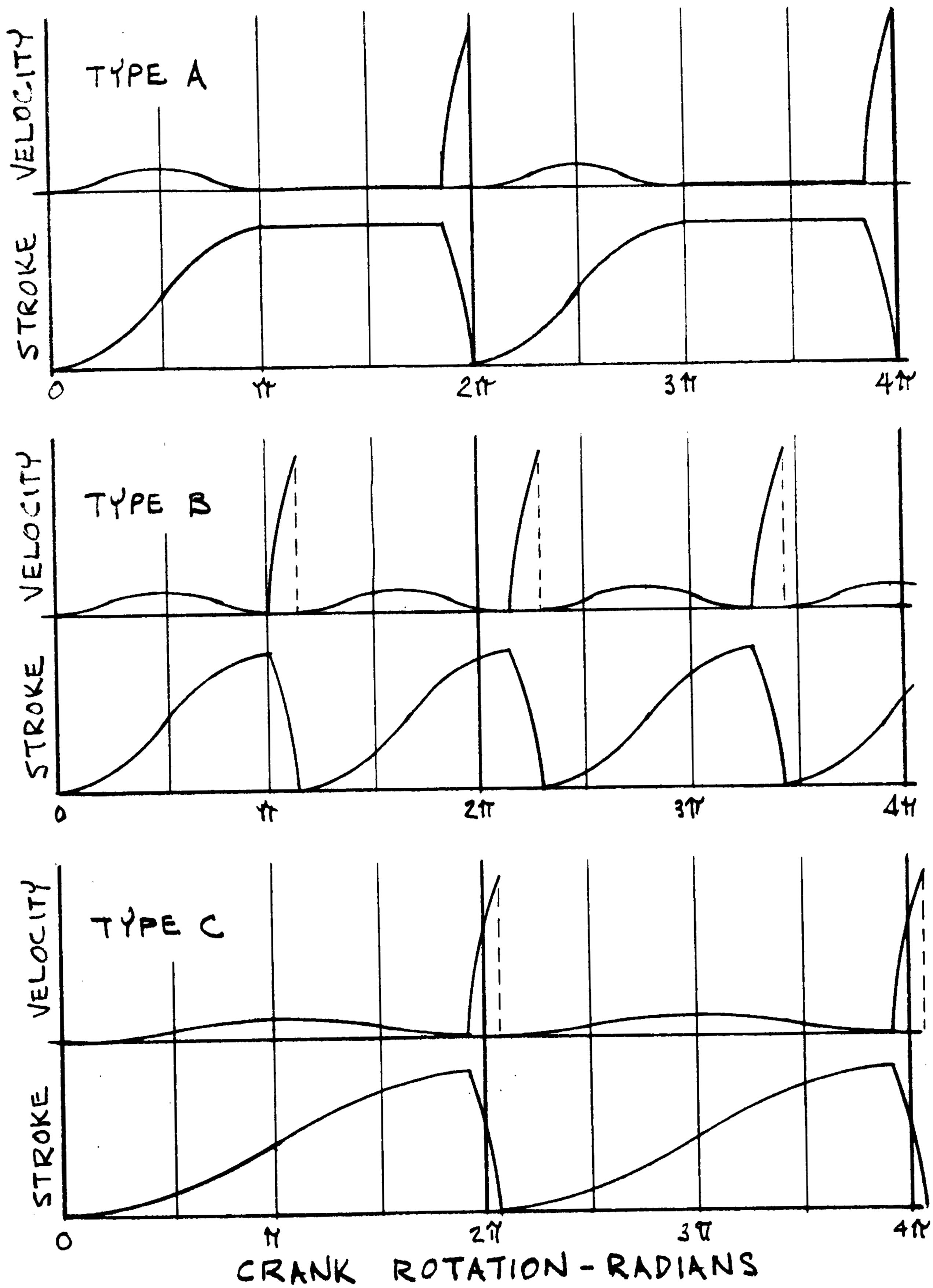


FIGURE 2 IMPACT HAMMER CHARACTERISTICS

peak hammer velocity during the retraction part of the stroke is the same for type A and B and is slightly less than 1 fps. Since type C uses a full rotation of the drive shaft to accomplish the hammer retraction, the peak retraction velocity is half that for types A and B. If the drive shaft velocity for type B is reduced to produce the same impact rate as type A and C, then the peak retraction velocity is also reduced to a value more nearly like that for the type C hammer. Thus, it would appear that the type B hammer is potentially more efficient than type A or C; however the primary influence on hammer efficiency is the reduction or elimination of sliding friction between the hammer and the guides. This suggests the use of some sort of rolling guide for the hammer which will be examined more fully as the design progresses.

Since a lighter hammer might be desired, it is necessary to examine the effect of varying this parameter. The velocity of the hammer while it is being accelerated by the spring is given by the relation

$$v^2 = \frac{kg}{W}(x_2^2 - x^2)$$

where W is the hammer weight, k is the spring constant, x_2 is the maximum compression of the spring, and x is some point in the stroke. Substituting this velocity into the expression for kinetic energy yields

$$KE = \frac{1}{2}\frac{W}{g}v^2 = \frac{W}{2g}\cdot\frac{Kg}{W}(x_2^2 - x^2) = \frac{K}{2}(x_2^2 - x^2).$$

From this it is seen that the mass of the hammer can be reduced without changing the kinetic energy in the hammer at impact; however, the velocity characteristic of the hammer is altered, as is the duration of the impact stroke. Since the spring driving force varies between 30 and 50 pounds, a good estimate of the time variation of the impact stroke can be achieved by assuming an average value for the acceleration of the hammer. This acceleration is given by

$$a = \frac{F}{W} g$$

where F is the spring driving force. The time required to travel a distance, s, is given by

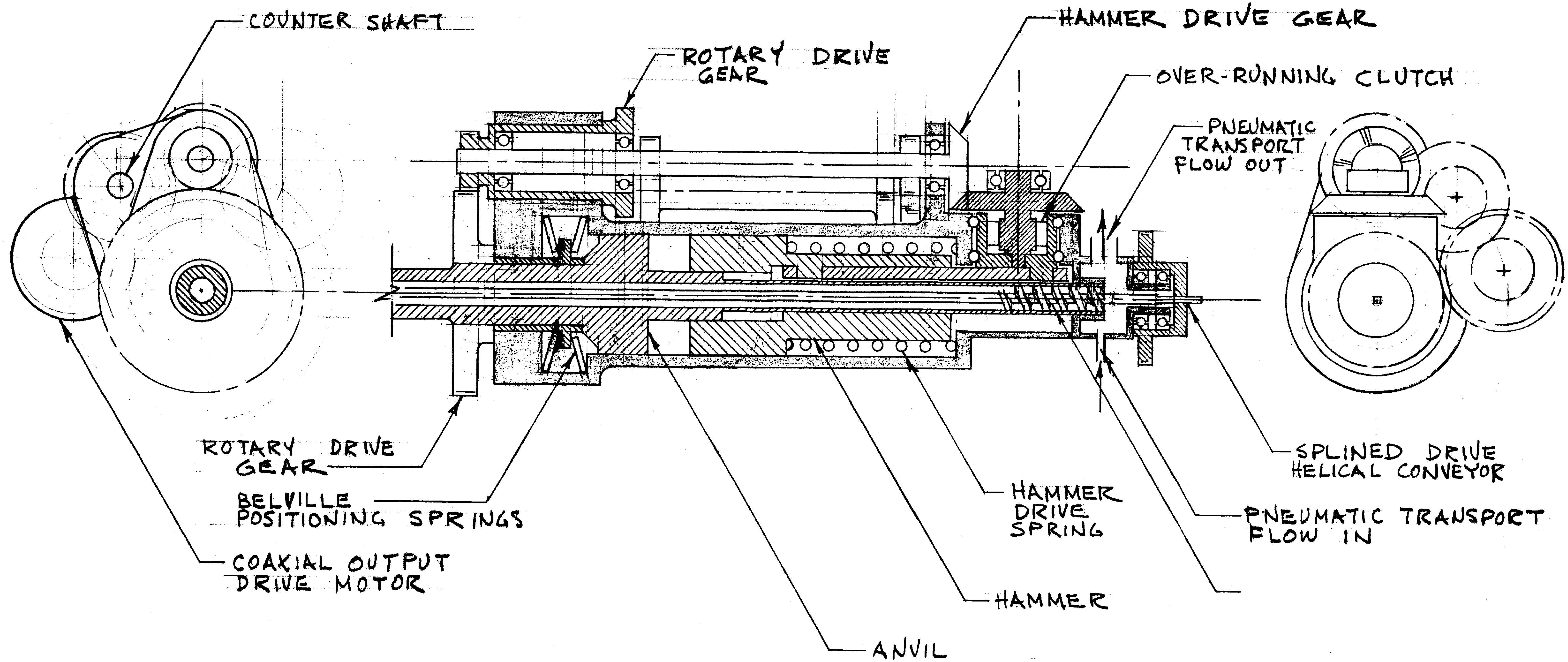
$$t^2 = \frac{2s}{a} = \frac{2sW}{Fg} \quad \text{and} \quad t = \left(\frac{2sW}{Fg}\right)^{\frac{1}{2}} .$$

Thus, it is seen that the duration of the impact stroke is reduced in proportion to the square root of the hammer weight. A reduction in hammer weight by a factor of four reduces the duration of the impact stroke by a factor of two. For this reduction in hammer weight, the type B hammer now produces impacts at the rate of 1.84 to one, which is approaching twice the rate of the type C hammer for the same driving speed, or conversely, almost half the input drive shaft speed for the same impact rate. This weight reduction will yield a hammer weighing slightly less than one-half pound which is more consistent with a light weight drill design.

A potential advantage of the type B hammer is that the over-running clutch will allow the hammer to rebound from the anvil partially compressing the spring. By virtue of the one way or ratchet type action of the clutch, this compression can be retained in the spring. Thus, some energy recovery is possible with this system. Since the rebound velocity of the hammer is determined by the relative mass of the hammer and drill, this feature of energy recovery could be important in a light hammer system in improving the efficiency of the system. A preliminary mechanization of the type B hammer is shown in Figure 3. This approach allows one motor to be used to drive the drill in rotation, operate the hammer, and to drive the helical conveyor. The over-running clutch in the impact hammer drive will also provide the capability of operation in two modes. One is a combined rotary/impact mode of drilling which is the normal mode for hard rock. By reversing the drive motor rotation, it is possible to run the drill in a simple rotary drilling mode since the over-running clutch will not now engage the hammer actuation crank. For the configuration as shown in Figure 3, reversing the drive motor will also reverse the helical conveyor rotation. Thus, if the dual mode of operation is desired, some means of providing the proper rotation of the helical conveyor is necessary. This can be accomplished by using a dual gear train drive to the helical conveyor having outputs of opposite rotational sense. The proper output is then engaged to the helical conveyor through a pair of simple over-running clutches.

As shown in the configuration of Figure 3, the sample is transported up the helical conveyor and empties into a closed chamber. A jet of air or compressed gas is introduced into the chamber which picks up and carries the sample pneumatically by the flow out of the chamber. The sample is carried through a tube to a cyclone collector located in the payload. While this approach is simple to mechanize, it suffers from the fact that a stored gas supply must be available. If the drill is to be used in a large number of repeated or prolonged operations, the amount of gas required could become excessively large. More efficient use of the pneumatic gas supply can be achieved if a valving system is incorporated with the chamber located at the top of the helical conveyor so that the flow is intermittent; i.e., flows occur for only a small part of each rotation of the drill. A completely mechanical sample transport system, while more complex, is probably more desirable for this sampler. Such a system will be pursued in the final design of this sampler mechanism.

Two methods were considered for deploying this drill. The first method is a sliding bar arrangement as shown schematically in Figure 4. In this arrangement the intermediate sliding bar is extended at the same time



IMPACT HAMMER/ROTARY DRIVE CONCEPT
FIGURE 3

FOLDOUT FRAME 1

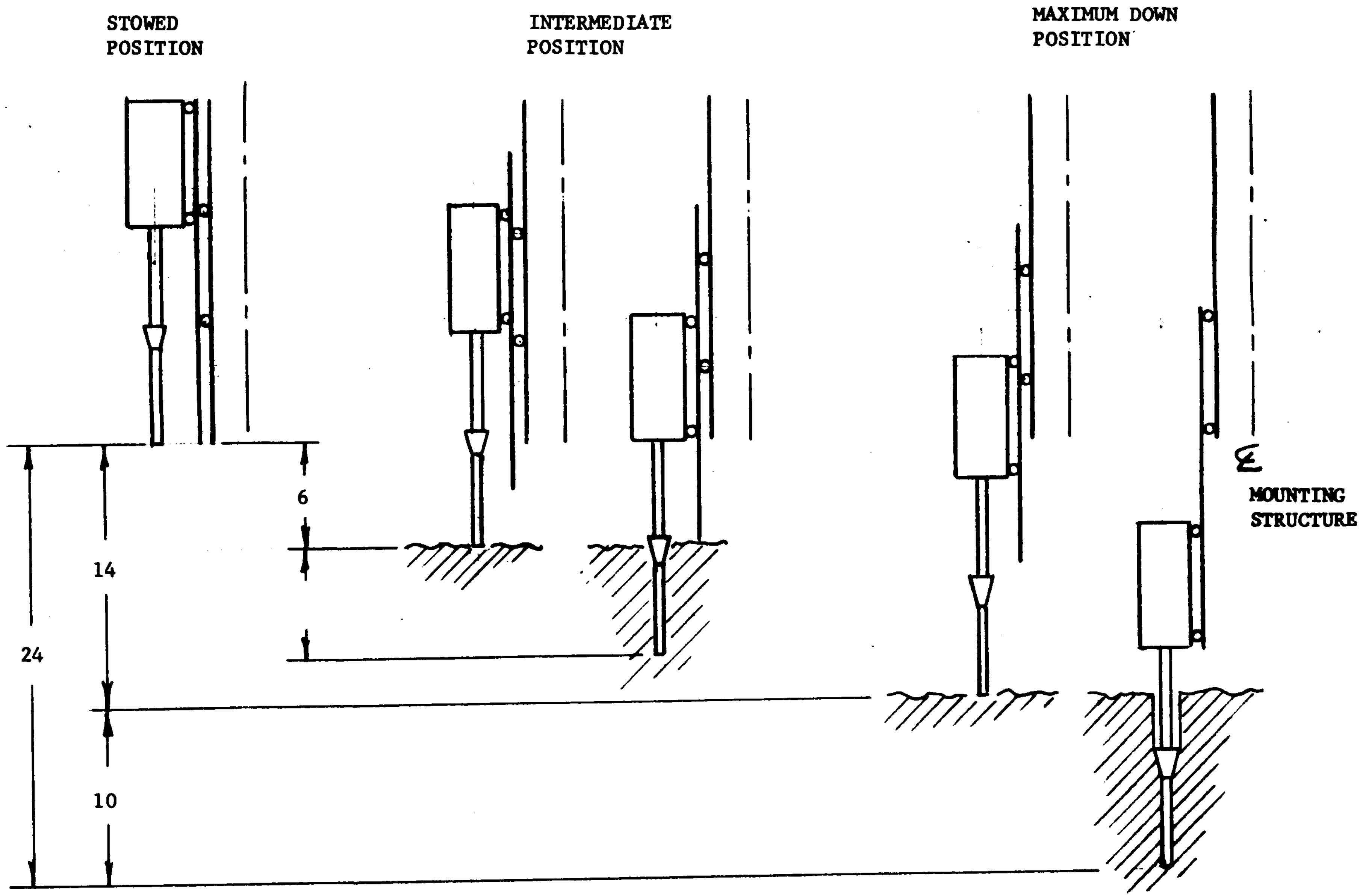


FIGURE 4. SLIDING BAR DEPLOYMENT MECHANISM

that the drill is traversing along it. The initial intent was to utilize a single drive for all the elements of the deployment mechanism. A defect of this scheme is that if the surface of the soil is encountered at some distance less than 12 inches below the stowed position, the depth of penetration is limited to a value equal to the distance traveled in reaching the surface. This is caused by the intermediate sliding bar reaching the surface which will then preclude further advance of the drill. This can be avoided by moving the drill and then the intermediate bar sequentially at the expense of some additional complexity in the drive mechanism. The maximum extension for this approach is 24 inches. Initial attempts to mechanize this approach did not result in as simple and compact a structure as appeared to be probable.

An alternate method for deploying the drill is shown schematically in Figure 5. This mechanism consists of two parts, a parallel bar linkage to deploy the drill to the surface and a longitudinal feed screw to advance the drill into the surface. The diagonal strut attached to the lower parallel bar link is an extendable member such as a lead screw running in a threaded sleeve. In operation the diagonal strut extends causing the parallel bar linkage to deploy the drill to the surface. When contact with a substantial surface is made, the deployment load builds up which can be sensed to terminate the deployment operation. The diagonal strut and lower parallel bar link then forms a rigid truss in conjunction with the support structure. The axial feed screw is then activated to begin the drilling operation. This mechanism can be stowed in fundamentally the same volume as the first approach; however, it provides more versatility in emplacing the drill both radially and vertically. Eight inches more vertical reach is achieved than with the first approach. One possible disadvantage is that this deployment mechanism could be less rigid; however, since the impact energy is low (approximately 1.5 foot pounds) and the axial thrust is low (approximately 20 pounds), the requisite rigidity should be achievable.

3.2 CASED ROTARY/IMPACT DRILL, E-2

No specific design effort has been pursued on this sampler at the request of JPL pending an ultimate decision on the desirability of such a device. However, it is noted that most of the design accomplished on the uncased rotary/impact drill (E-1) is directly applicable to the design of this device. The same impact hammer drive, helical conveyor sample transport, and deployment system can be used. In addition to the drill rotary drive, the casing must be capable of being driven independently in both rotation and axial feed. The intent of this sampler concept is to provide the capability of penetrating a loose overburden until solid rock is reached. At This point the casing is to effect a seal, where the hole enters the rock, to prevent contamination of the sample collected from the rock by material contained in the overburden. The additional mechanical complexity of this device is felt to be difficult to justify on the basis of the probability of encountering solid rock near the surface and the ultimately lower reliability associated with a more complex operation.

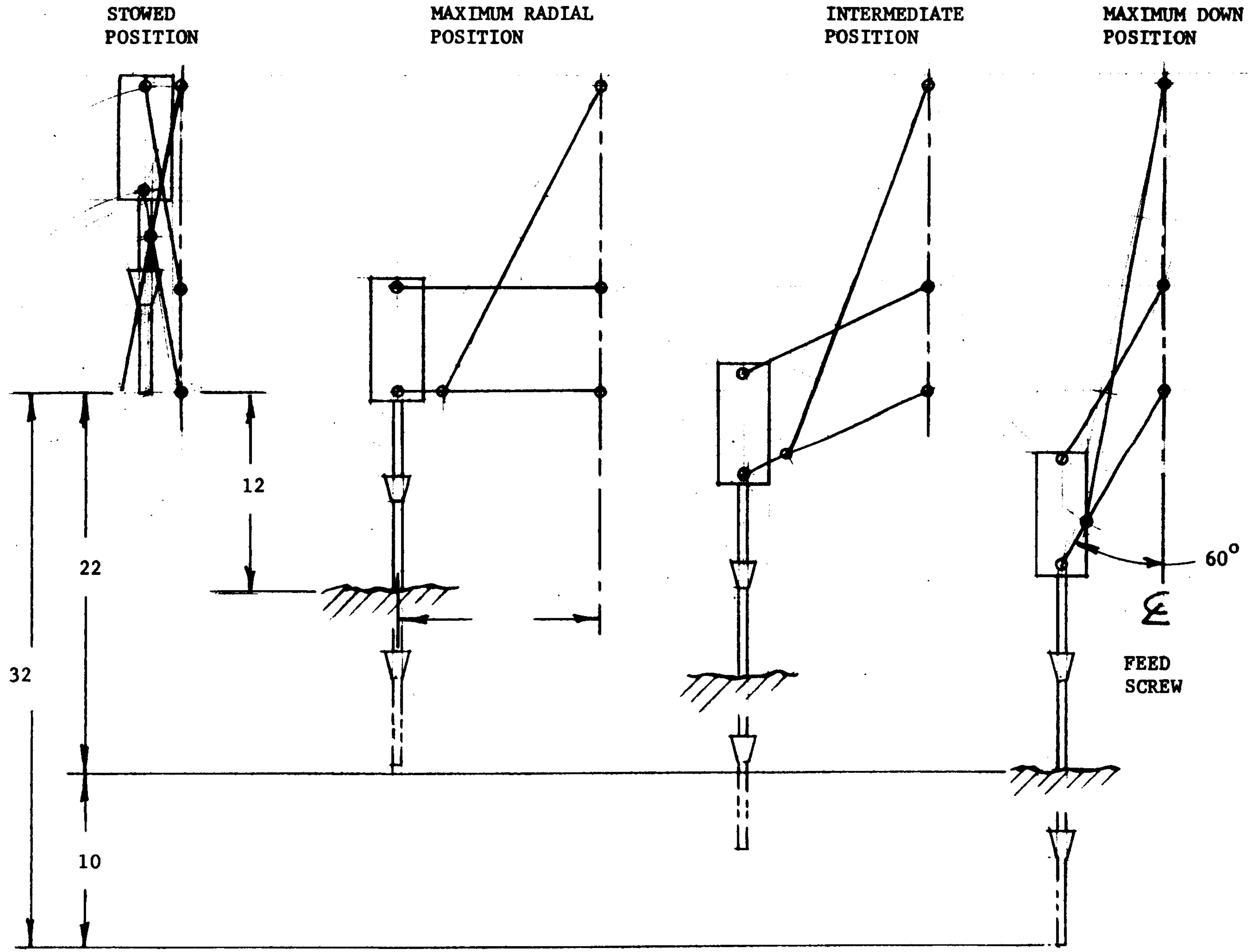


FIGURE 5. PARALLEL BAR DEPLOYMENT MECHANISM

3.3 CONICAL ABRADING SIEVE CONE, E-3

This sampler design effort is based on the deep abrading cone sampler developed by JPL and tested in the field as reported in the final report covering that task. In the form tested, this sampler used a helical conveyor running in a rubber lined casing. Since this method of soil transport was used on the uncased rotary/impact drill design (E-1), it was decided to explore other approaches for transporting the sample with this sampler mechanism. No changes in the external configuration of the abrading cone were made in this design since the existing configuration performed satisfactorily in the field tests.

Two sample transport methods using pneumatic transport were considered. The first approach is shown in Figure 6. In this approach the abraded soil sample passes through the slotted openings in the cone and collect at the base. When sufficient sample has collected to a depth which covers the base of the low speed helical conveyor or auger feed, soil is carried up the short helical conveyor and spills into a closed chamber. The drive shaft to the helical conveyor is also a tube that provides a path for pressurized transport gas which emanates from the exit tubes creating an aerosolizing jet to pick up the soil particles. The flow out of the chamber up the drive shaft of the abrading cone carries the soil sample through a transfer tube to a cyclone collector. The soil in the feed auger or helical conveyor acts as a seal to prevent the pneumatic flow along the auger. The gas supply is valved intermittently to conserve the amount of transport gas required.

Another approach based on the valved sample transport system used on the breadboard model of the cylindrical abrading sieve with closed pneumatic transport developed by JPL is shown in Figure 7. In this case an internal cavity attached to a hollow shaft mounted concentrically with the abrading cone drive shaft serves as a receptacle into which the soil is deposited. This central tube is fixed to the support structure so that it does not rotate with the abrading cone drive shaft. The soil particles enter the conical head through the entry slots and fall to the bottom of the cone where it is free to enter the cavity through an opening in the bottom third of the internal cavity. An elastomeric seal bonded to the inside surface of the abrading cone closes this opening once in every rotation of the abrading cone, since the internal cavity does not rotate. At the same time a valve located at the top of the abrading cone admits a quantity of pressurized gas which emerges as an aerosolizing jet from the gas outlet tube. This gas then flows up the central support tube to a cyclone collector carrying the soil sample with it.

The breadboard model tested in the field had the drive and feed mechanism gimballed at the upper end so that the abrading head could swing away from a nominal vertical position to which it was spring loaded. This was

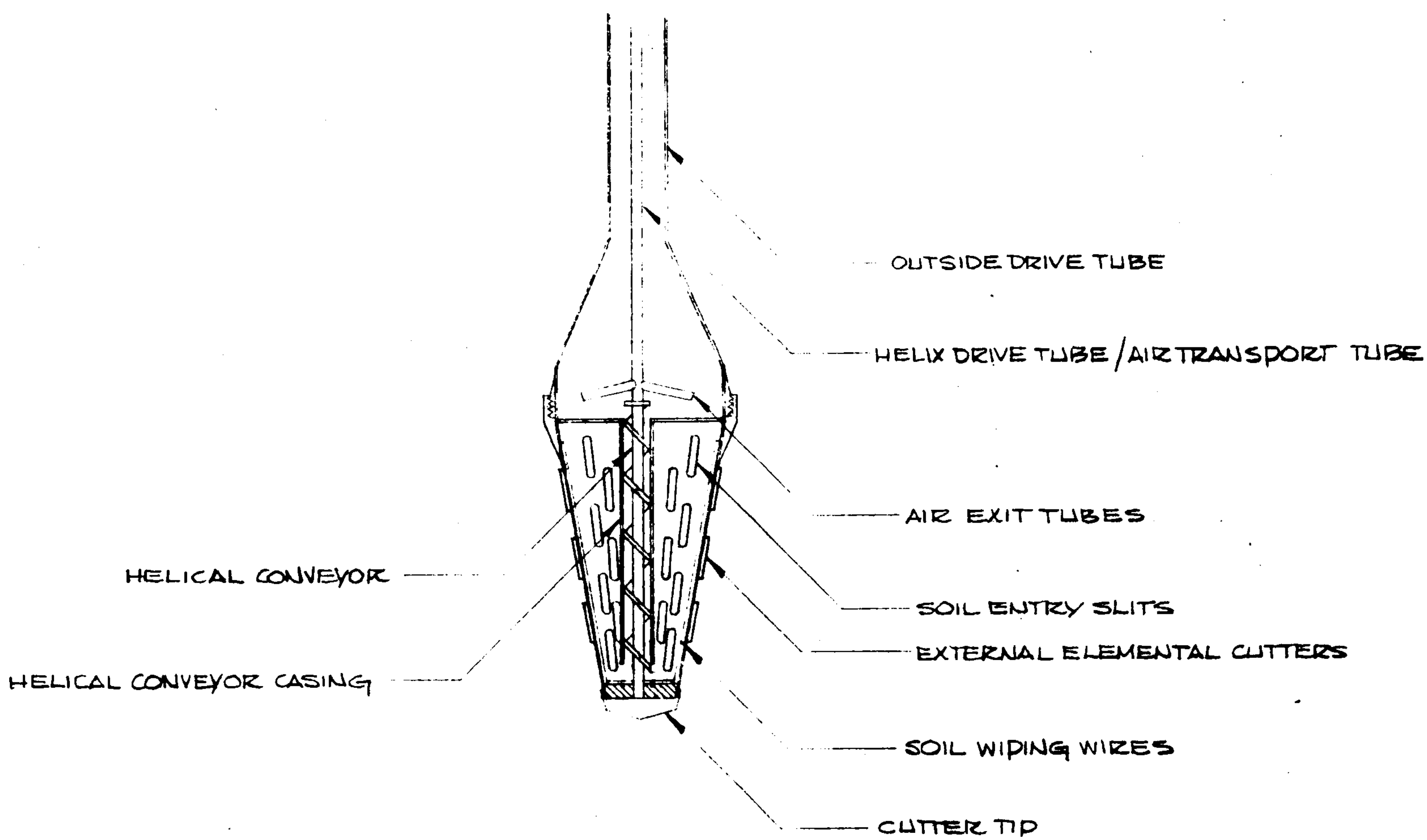


FIGURE 6. AUGER FED PNEUMATIC TRANSPORT

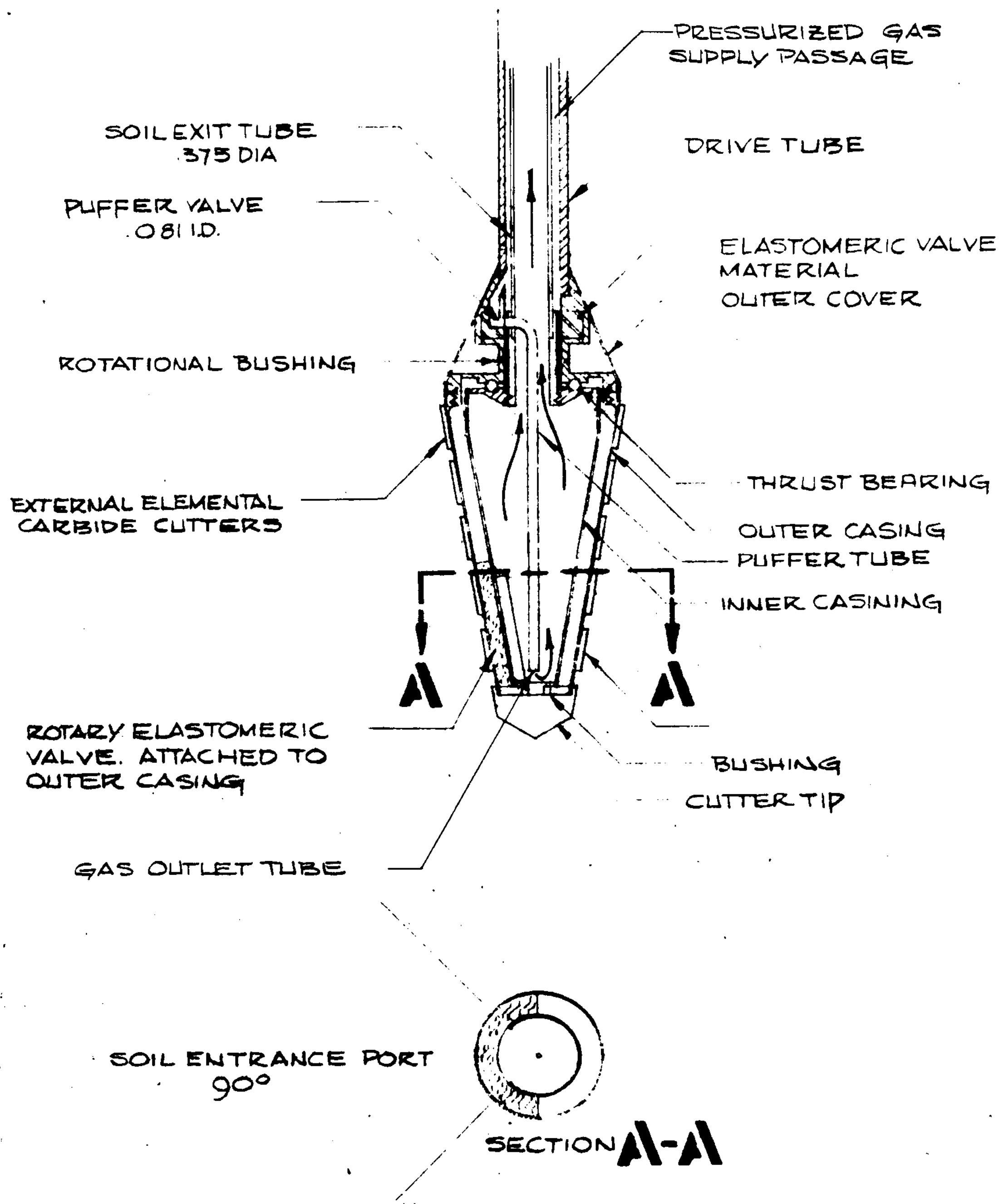


FIGURE 7. VALVED PNEUMATIC TRANSPORT

done to allow the abrading head the freedom to seek a desirable entry point into the surface between any rocks or cobbles that might be encountered. This requires the feed to be also gimballed so that the axial feed is always applied in a direction parallel to the axis of the abrading cone and the hole it digs. The current design approach is to replace the four bar linkage and pivot which provided this freedom of movement with a spherical ball joint which should be more compact. Cantilever springs arranged around the periphery of the drive mechanism will serve to spring load the mechanism to the nominal vertical position. These springs will also react the torque generated by the abrading head.

It is planned to use a parallel bar linkage to deploy the sampler from the stowed position to the surface in a manner similar to that used by the uncased rotary/impact drill (E-1). A modified version of the canted roller feed mechanism, used on the VCS sampler developed by Philco-Ford, was considered first for obtaining the required feed; however, this approach did not lend itself to a clean solution for obtaining a pneumatic transport path past the feed mechanism. Alternative approaches being considered are a lead screw feed and cable actuated telescoping tubes mounted concentrically around the axis of the sampler mechanism.

3.4 HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER, E-4

This design is based on a JPL prototype using a helical conveyor mounted on a flexible shaft so that it can be run in a curved housing. This approach differs from the simple helical conveyor tested in the field in this respect. The model tested in the field had a rigid helical conveyor running in a straight casing or housing. No rubber liner is used in the housing for this conveyor, although it could be considered.

The prototype model built by JPL has two bends in the helical conveyor as shown in Figure 8. The housing is rotated about the axis as shown so that the tip of the conveyor sweeps through the arc R_1 and enters the surface of the soil. The first 5 to 6 inches of the conveyor housing is formed to a radius equal to R_1 so that it can follow the tip through the curved hole made in the soil without interference. In loose material the sampler can proceed past this point but in cemented material the penetration of the sampler is limited to this 5 or 6 inch length. The radii R_2 and R_3 are determined by the flexibility of the helical conveyor shaft. These should be kept as large as possible consistent with small size to minimize power lost in flexing the helical conveyor shaft as it rotates. The approach taken in this design is to eliminate the bend associated with the radius R_3 while retaining the ability to deliver the sample to a fixed point. Such a configuration is shown in Figure 9. In this configuration the exit point is located over the receiving cup in such a manner that it sweeps across the cup symmetrically with respect to the vertical center line of the rotation point as the sampling tip

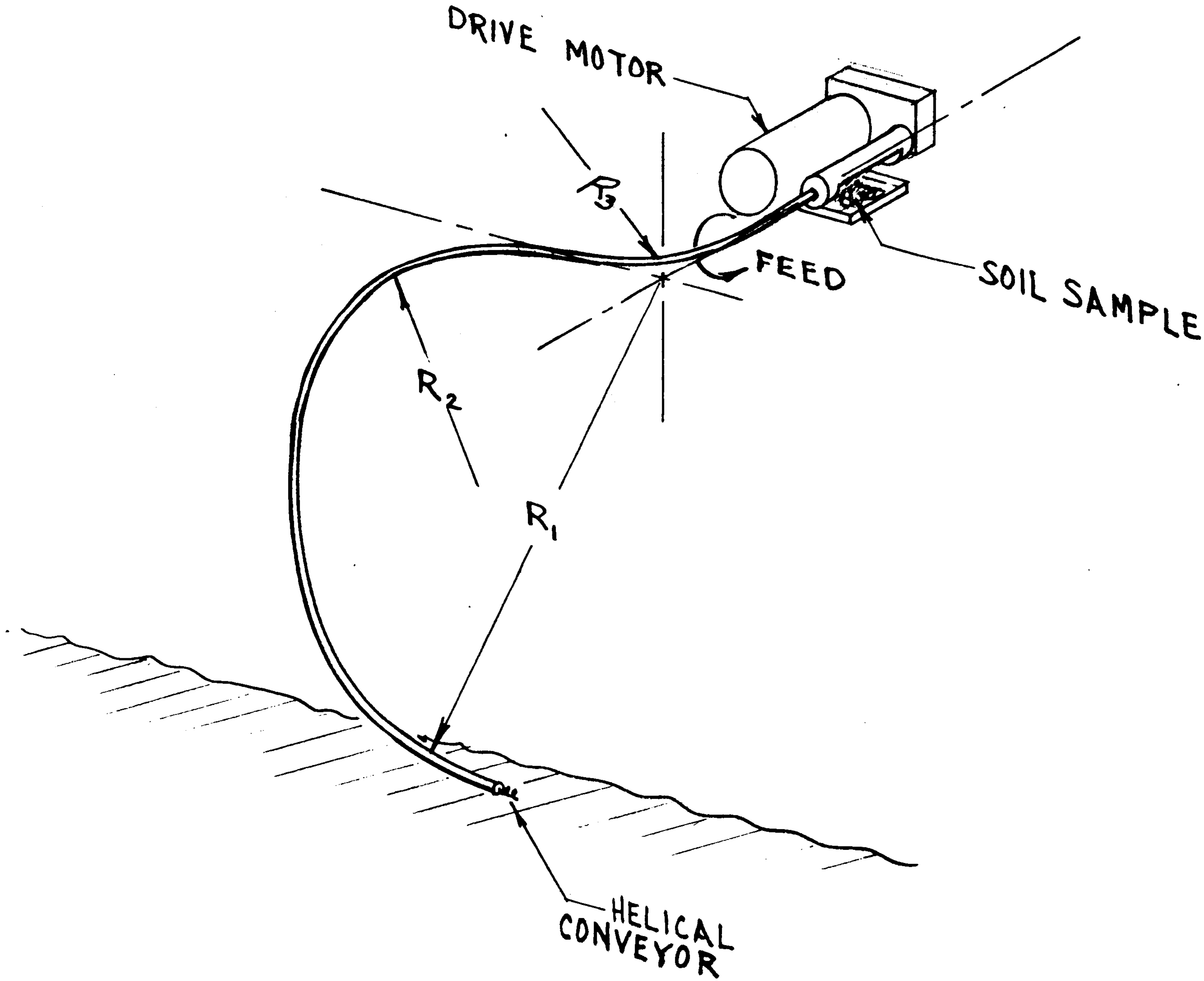


FIGURE 8. JPL PROTOTYPE GEOMETRY - HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER

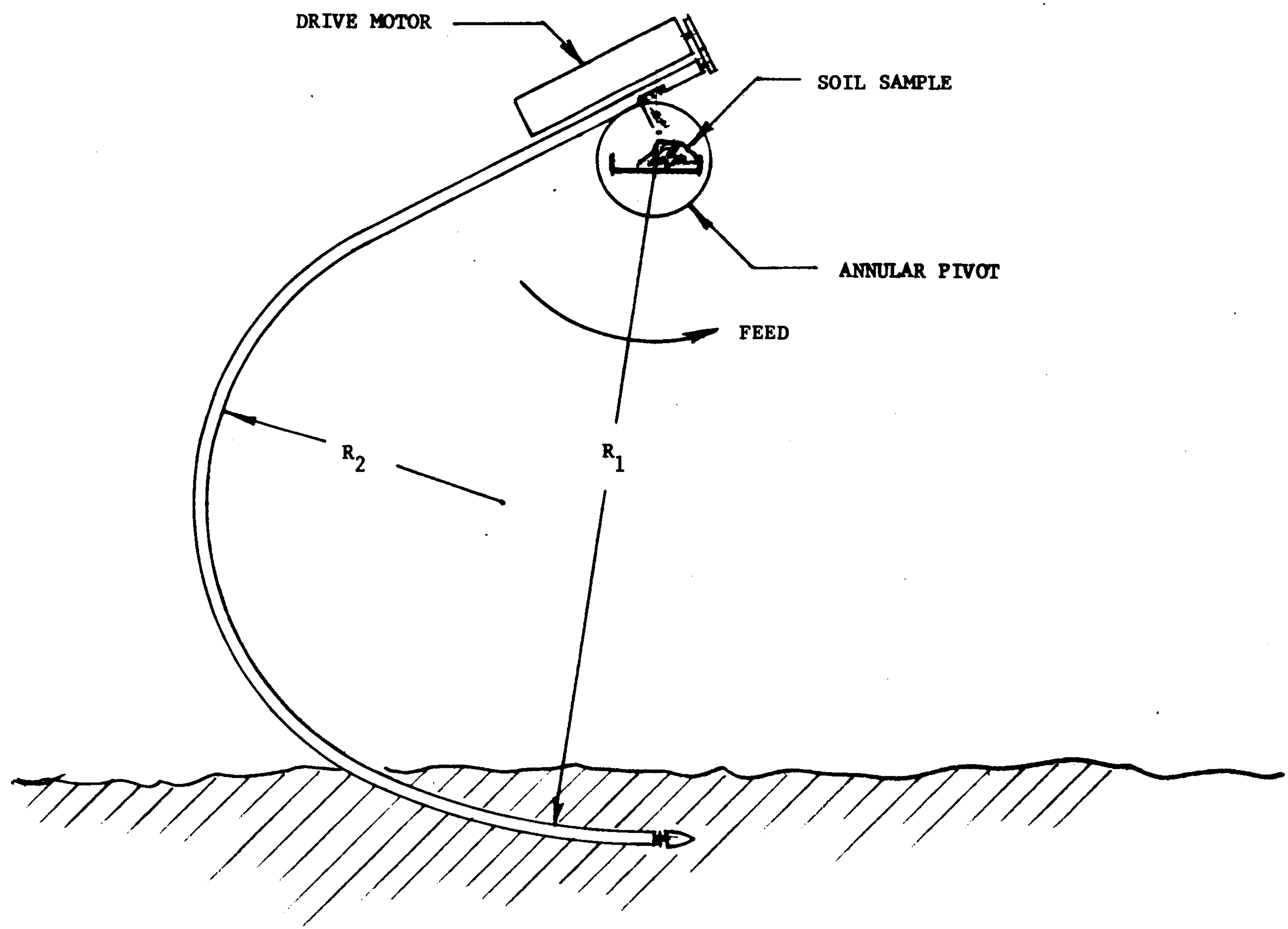


FIGURE 9. SINGLE BEND HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER

traverses through a level surface. This necessitates the use of an annular bearing or support point so that the receiving cup is located at the center of rotation.

The feed of the sampler is provided by a geared drive through a slip clutch set to limit the axial thrust to a value compatible with the strength of the conveyor housing. This axial thrust at the sampler tip is designed to be limited to 2 pounds maximum.

The single bend simple particulate sampler should require less power to operate and will have a much more compact operating envelope. It should also be structurally sounder and more resistant to damage because of the reduced length of the conveyor housing and the elimination of some of the eccentric loading.

3.5 BACKHOE SAMPLER, E-5

3.5.1 SCOOP CONFIGURATION

This sampler utilizes a scoop mounted on the end of an extendable boom which can be drawn across the surface to collect a sample. Repeated operation in the same location will allow trenching to some depth below the surface consistent with the strength of the soil surface being sampled. Design criterion number 1, which limits the maximum particle size to 10 millimeters, and criterion number 7, which provides that no rocks shall prevent complete closure of the scoop, are felt to be fundamental to the configuration and design of the scoop. A concept was generated in which a thin blade is mounted on the boom ahead of the scoop as shown in Figure 10.

The function of this blade is to minimize hang ups on large rocks by causing the scoop to ride up and over and to start intermediate size rocks moving to one side or the other so that they will tend to flow around the scoop. The scoop itself is configured so that the width at the inlet is compatible with the maximum size particle that is acceptable, i.e., 10 millimeters between the edge of the scoop and the divider blade. The cutting edge of the scoop is positioned so that it shears the soil at an angle of 30 degrees from the horizontal. The internal configuration of the scoop is such that the soil entering it rests on an essentially horizontal surface. Thus, if the scoop is lifted off the surface before it is closed, the soil sample will not all fall out.

In order to determine the positioning requirements for the scoop, the geometry of the boom relative to the surface being sampled must be considered. This geometry is shown schematically in Figure 11. The length of the boom, ℓ, the height of the boom support, h, and the angle between the boom and the surface, α, are related to each other by the expression

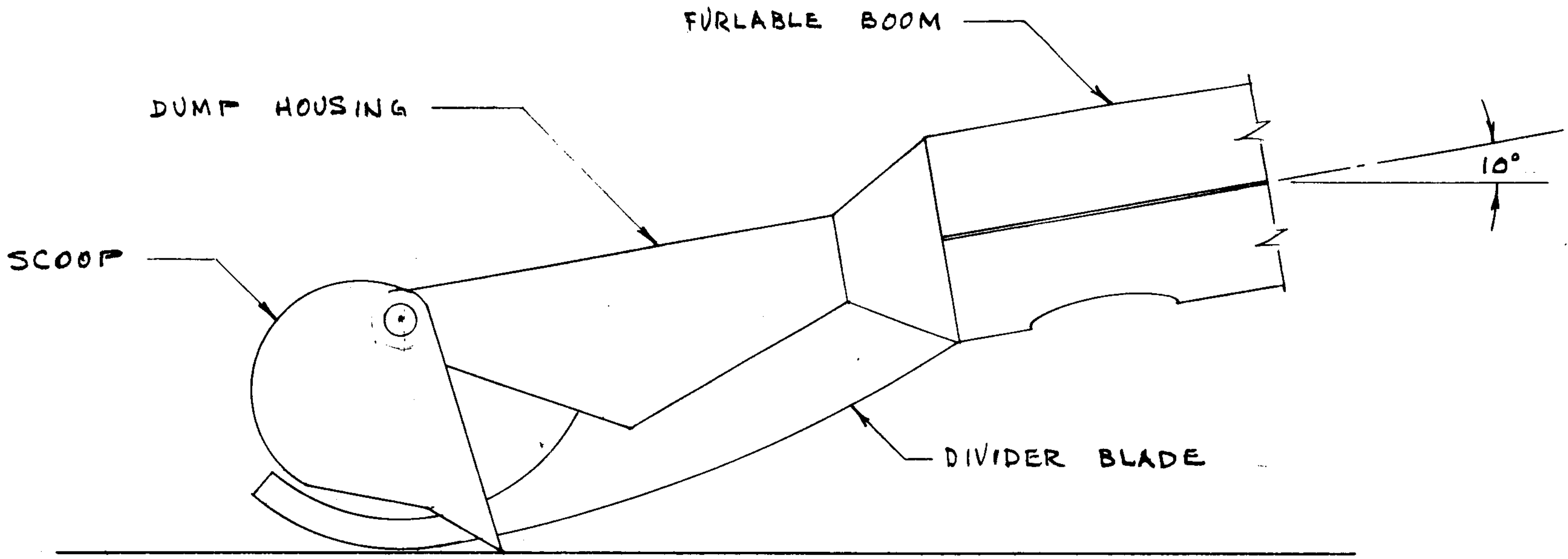


FIGURE 10. BACKHOE SCOOP CONCEPT

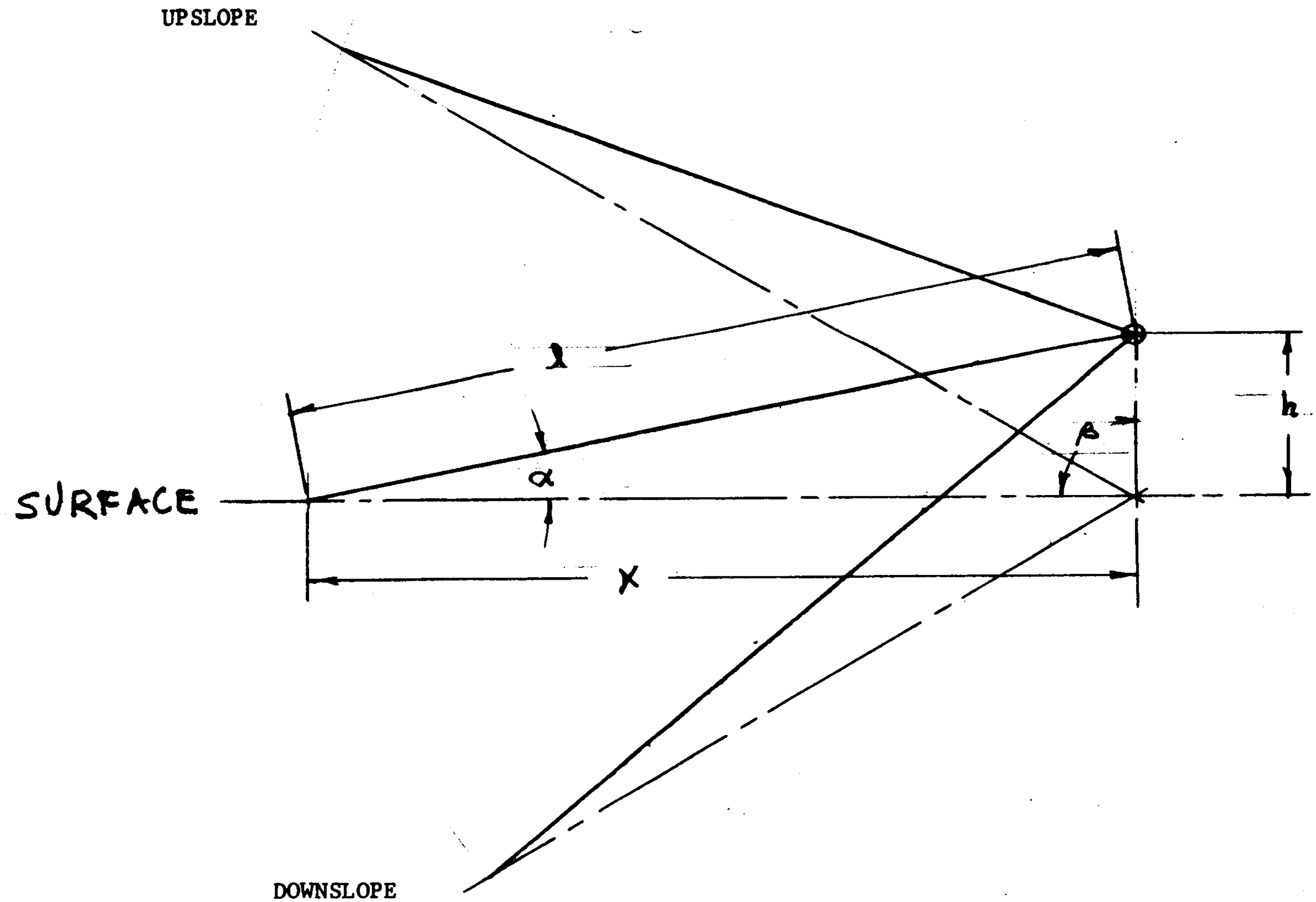


FIGURE 11. BOOM DEPLOYED BACKHOE GEOMETRY

$$\sin \alpha = \frac{h}{\ell} \sin \beta$$

It is assumed that the surface can lie anywhere between a 30 degree upslope and a 30 degree downslope from the boom support axis, the height of the support point varies from 12 to 48 inches, and that the maximum down angle of the boom is 60 degrees below the vertical.

The curves in Figures 12 and 13 show that the angle at the tip of the boom between the axis of the boom and the surface varies in essentially the same manner and magnitude regardless of the slope of the surface, at least between a 30 degree upslope and a 30 degree downslope. It also indicates that the angle increases steadily as the boom is retracted to a maximum value of 60 degrees on a level surface and 90 degrees on a 30 degree downslope when the boom is in its most depressed condition. Figure 14 indicates the minimum length of boom required to reach the surface as a function of support height, h. Thus, a five foot long boom must be mounted less than 3 feet above the surface in order to collect a sample. A boom length of 10 feet was assumed initially for this sampler.

Based on the large variation in the angle α, it is desirable that the scoop is always positioned in the same way with respect to the local vertical regardless of the position of the boom. This implies some sort of positional feedback and servo to continuously adjust the scoop relative to the boom as the sampling traverse progresses. Thus, the cutting edge of the blade attacks a level surface at a 30 degree angle. As sampling progresses the soil collects in the scoop.

After collecting the sample the scoop is closed so that the sample can be transported to the payload. Two methods can be used to perform the sample transfer. One is to simply position the scoop over a dump port or funnel in the payload and open the scoop dumping the sample. The other is to utilize the boom as a transport path by raising it to a vertical position and allow the sample to fall down the boom by the action of gravity. The latter approach was used in the design of the booms which will be described subsequently.

Two approaches for actuating the scoop were considered. The simplest to mechanize is to mount a motor at the tip of the boom as shown in Figure 15. This scheme requires positional feedback for the boom in order to servo drive the scoop to the correct attitude. With this approach powerful actuation forces are achieved through the worm gear drive. A disadvantage is that the weight of the motor is placed on the tip of the boom which increases the reaction moment required to raise the boom. For very light weight furlable booms, the increase in bending moment can be critical in terms of the strength of the boom.

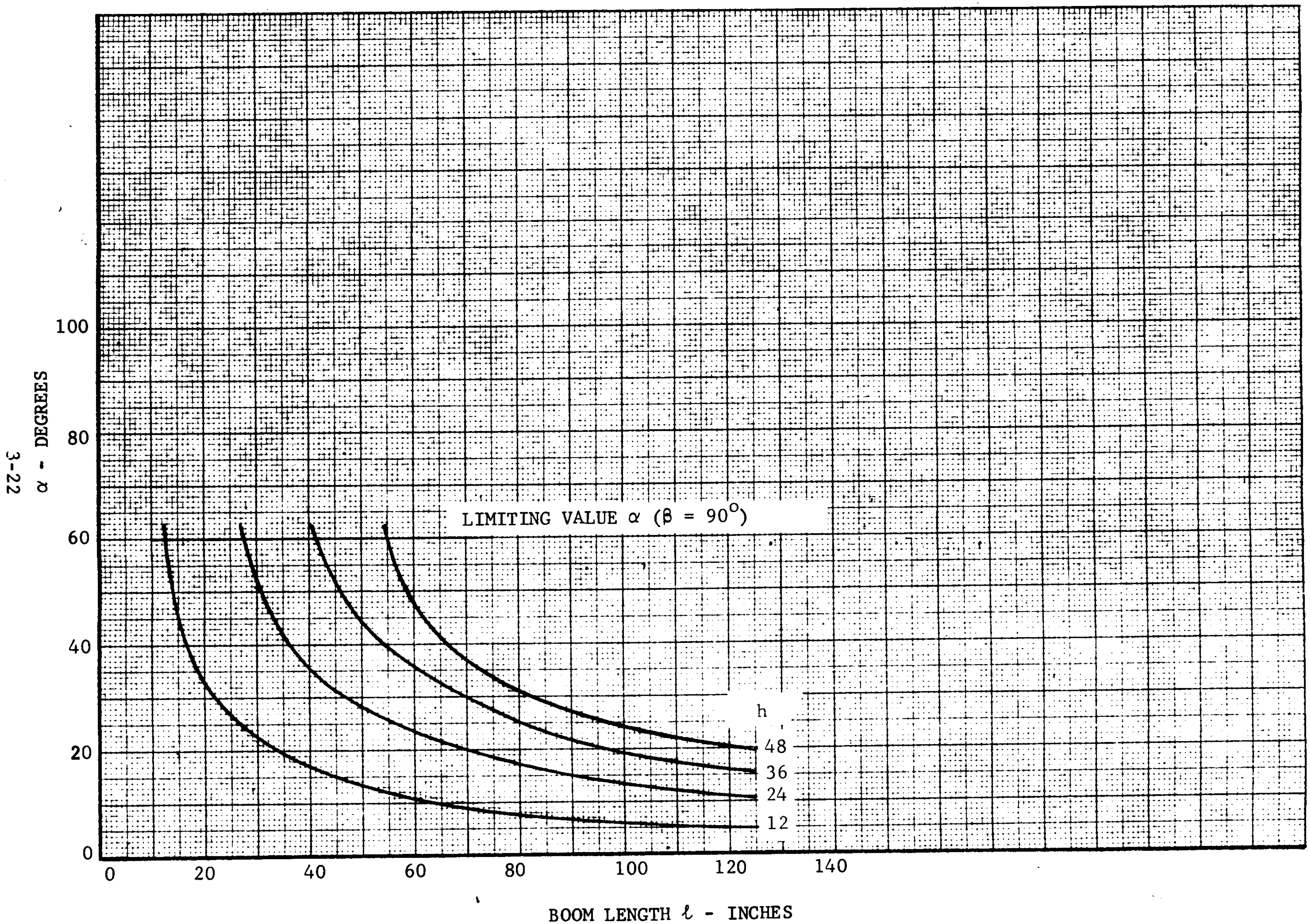


FIGURE 12. BOOM TIP ANGLE VARIATION WITH LENGTH ON A LEVEL SURFACE

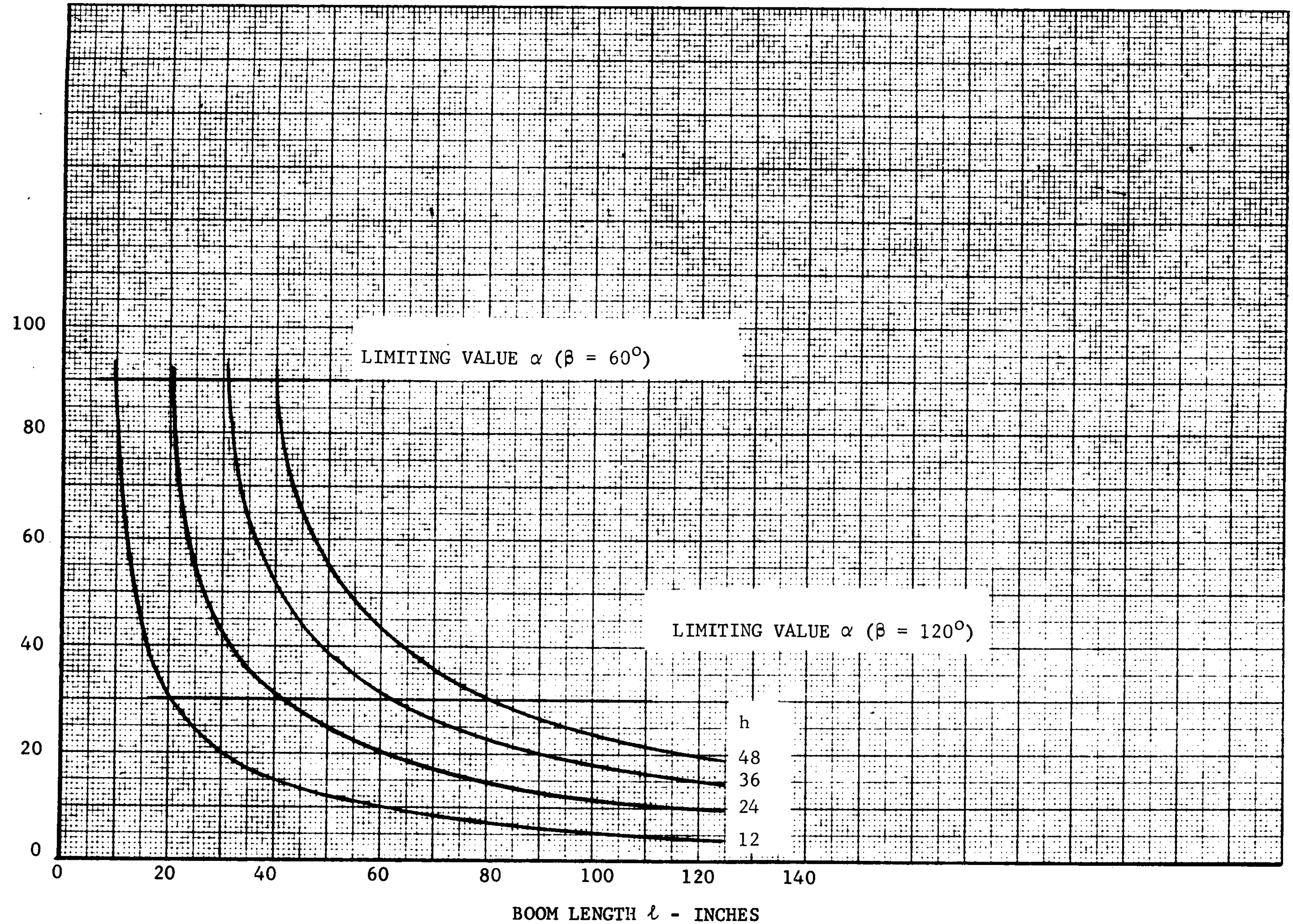


FIGURE 13. BOOM TIP ANGLE VARIATION ON A SLOPED SURFACE FOR $\beta = 60^\circ$ AND 120°

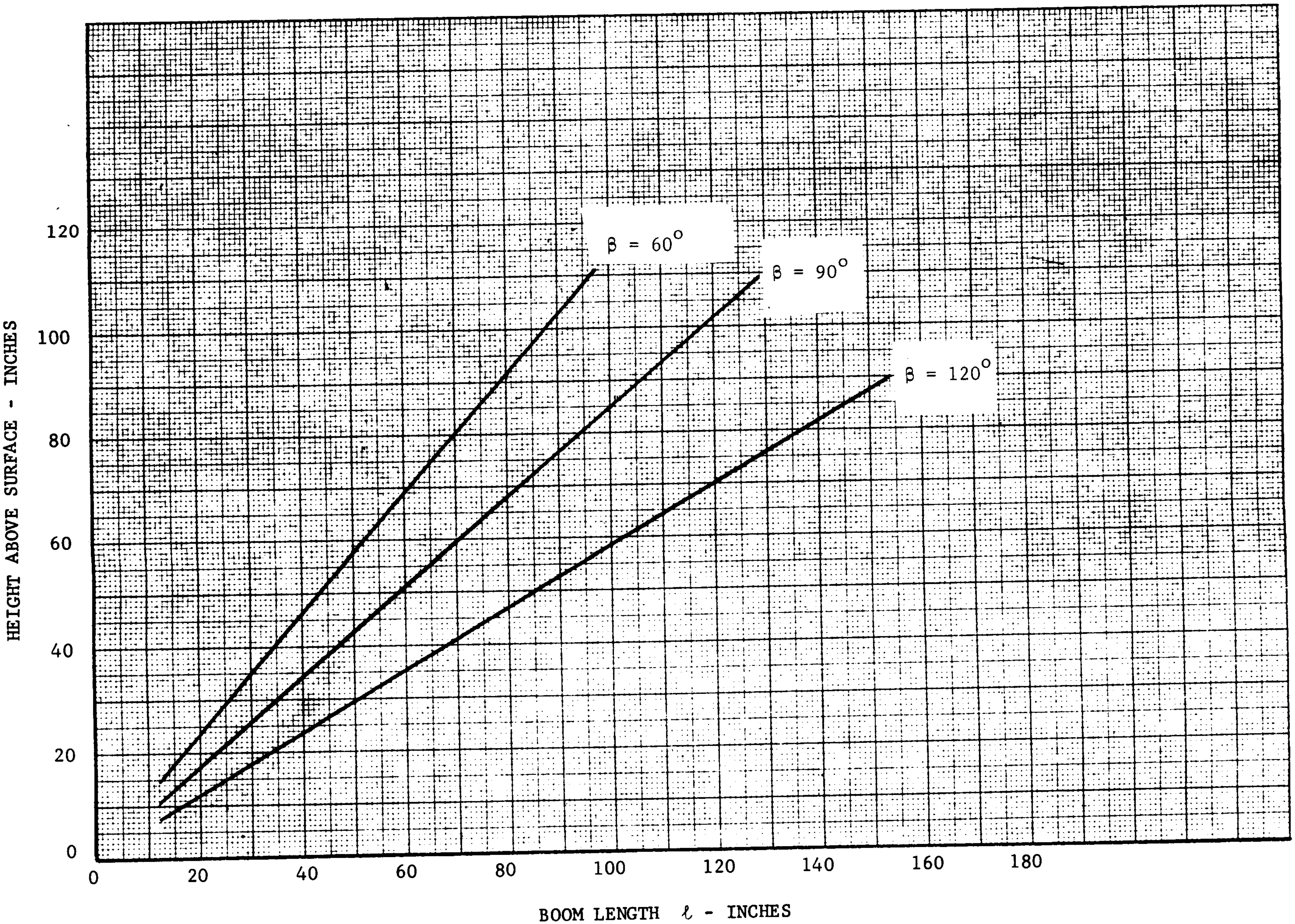


FIGURE 14. MINIMUM LENGTH OF BOOM TO REACH SURFACE AT LIMITS OF DOWN DEFLECTION OF BOOM

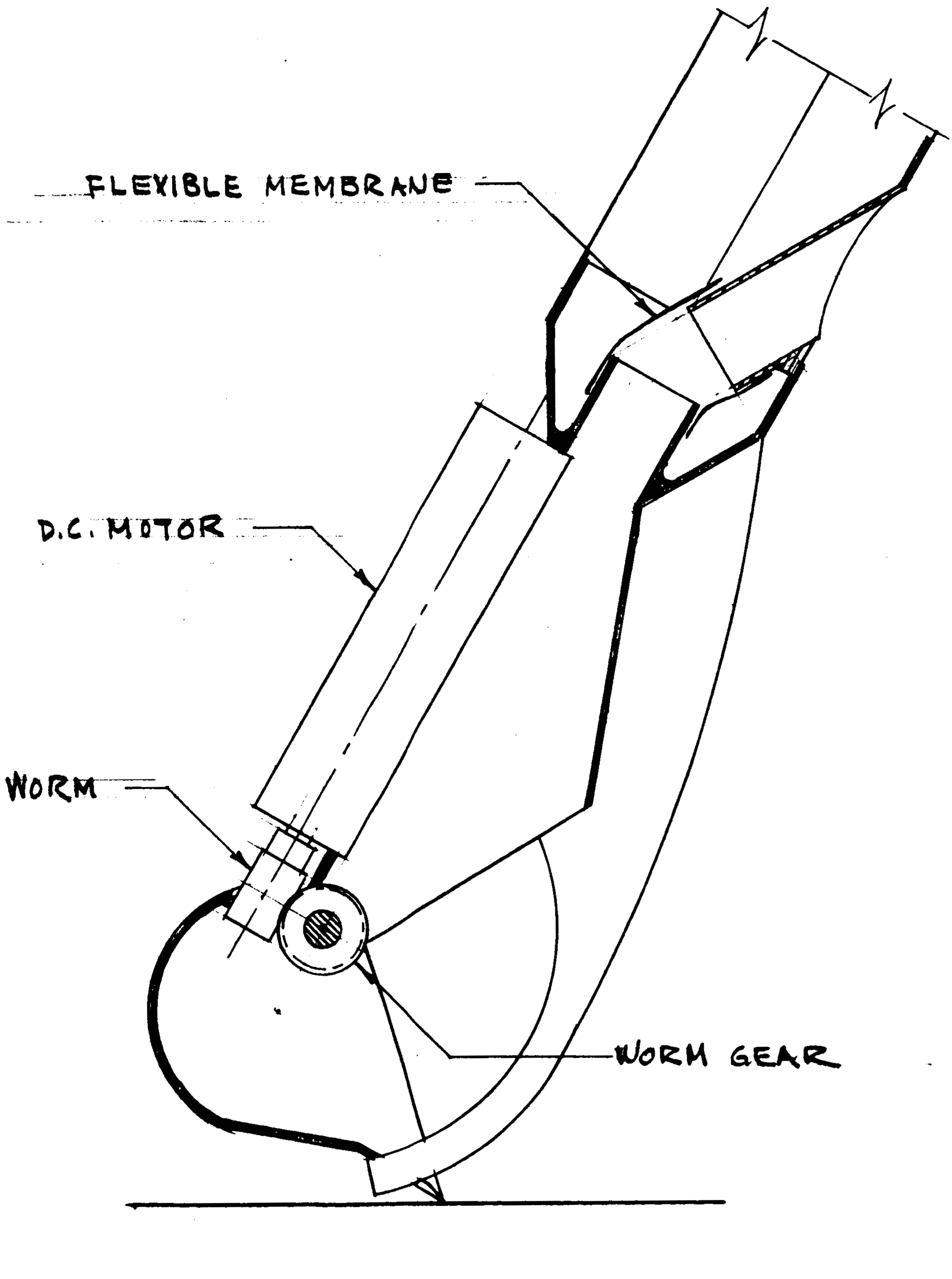


FIGURE 15. BACKHOE SCOOP - MOTOR ACTUATED

The alternate approach considered was a cable system which provides mechanical feedback and actuation forces. This configuration is shown in Figure 16. This approach uses parallel cables fixed to a drum on the scoop. The other ends are wrapped around another set of drums at the base of the boom. As the boom extends or retracts, these drums rotate in opposite directions to feed out or take up the appropriate amount of cable as dictated by the length of the boom. As the boom is elevated or depressed, these cables act in a manner similar to the bands on a drafting machine to maintain the scoop in the same orientation regardless of boom attitude. To open or close the scoop, the drums at the base of the boom are rotated in unison in the appropriate direction. While this system is somewhat more cumbersome, it places less weight on the end of the boom and automatically maintains the scoop in the proper position.

3.5.2 FURLABLE BOOM

An initial design approach was oriented towards using the gravity dump mode with the Ryan furlable boom. This boom has a closed cross-sectional shape which gives the boom more structural integrity than the DeHavilland boom (particularly in torsional rigidity) although not as much as can be achieved with a telescoping boom. Thus, some exit port for the sample must be incorporated at some point in the furlable boom that does not degrade its strength. Two possible locations are available. One near the tip of the boom in the section which is not flattened after complete retraction and the other at the base of the boom when it is fully deployed. This latter location for the exit port must be behind the boom support guides in the transition section. The disadvantages of locating the exit port at this point is that the opening in the tube must be compatible with flexing involved in retracting the boom. Also, the interior of the boom is exposed to soil particles throughout its entire length. Any residual material would seriously degrade the life of the boom and could result in local failure of the furlable boom. Another possible disadvantage is that the boom must be fully extended during the gravity dump cycle. All these disadvantages can be circumvented if the exit port is located near the tip where a minimum of flexing occurs and the bending stresses are minimized. Also, since no flexure occurs for this portion of the boom, an auxiliary small diameter tube can be mounted inside the furlable boom to act as a soil transport chute thus eliminating the chance that soil particles would be inside the boom when it is retracted.

The furlable boom developed by Ryan Aeronautical Company consists of two preformed thin titanium alloy sheets resistance welded together at the edges which forms a cross-sectional shape as shown in Figure 17.

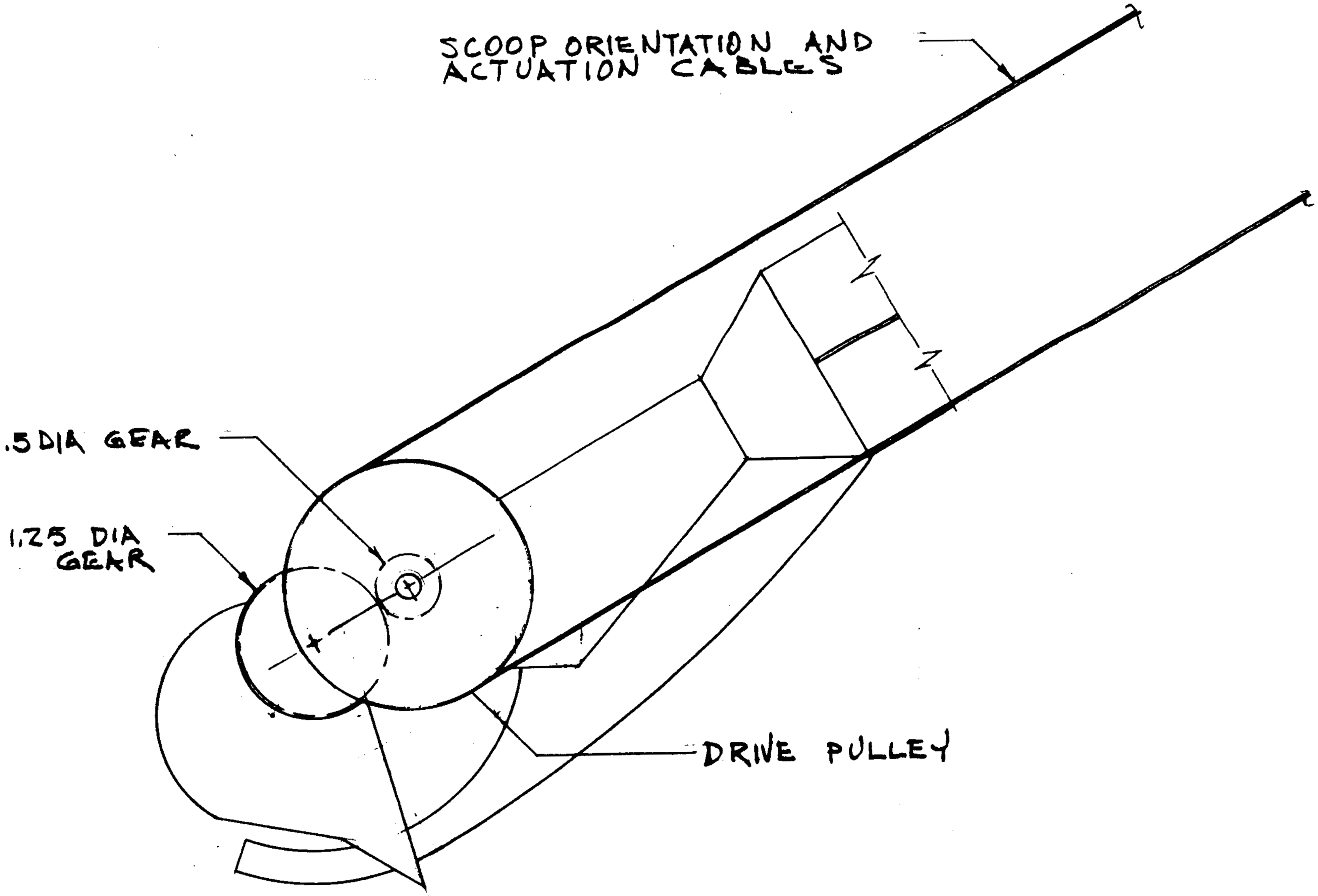


FIGURE 16. BACKHOE SCOOP CABLE ACTUATION SYSTEM

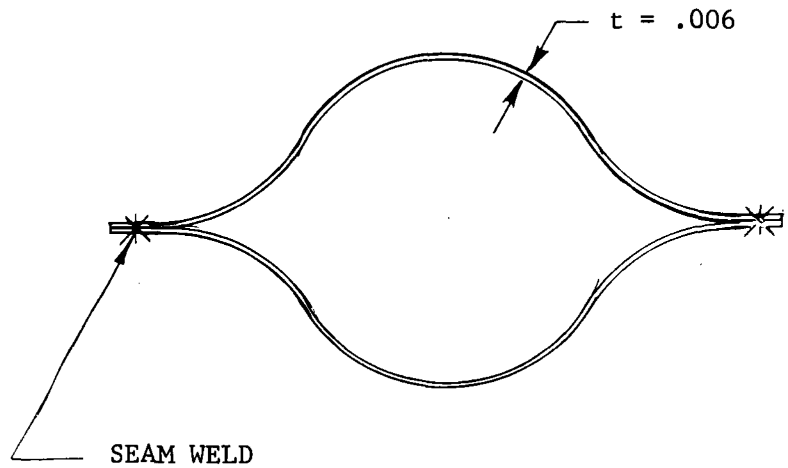


FIGURE 17. RYAN FURLABLE BOOM CROSS-SECTION

This boom is stowed by elastically flattening the tube and winding it onto a drum. The first consideration in using this boom to transport soil to the payload was to provide an opening in one side of the tube near the base when the boom was fully extended. When the boom was erected to the vertical the soil would fall down the inside of the tube and out the opening at the base into a receptacle. The disadvantages to this approach are as follows:

(a) Sample transport to the payload can only occur when the boom is fully extended.

(b) There is no assurance that the internal cleanliness of the boom can be maintained. Soil particles inside the boom would adversely affect the flexure characteristics during stowage on the take-up drum.

(c) The interface between the receiving receptacle and the exit hole in the boom is unduly complex.

Another approach to soil delivery with this boom was then considered and is shown schematically in Figure 18. In this approach the boom must be fully retracted to deliver a soil sample which at least eliminates extending the boom again after making a sampling attempt. The opening is now located near the tip of the boom where bending stresses are low and elastic

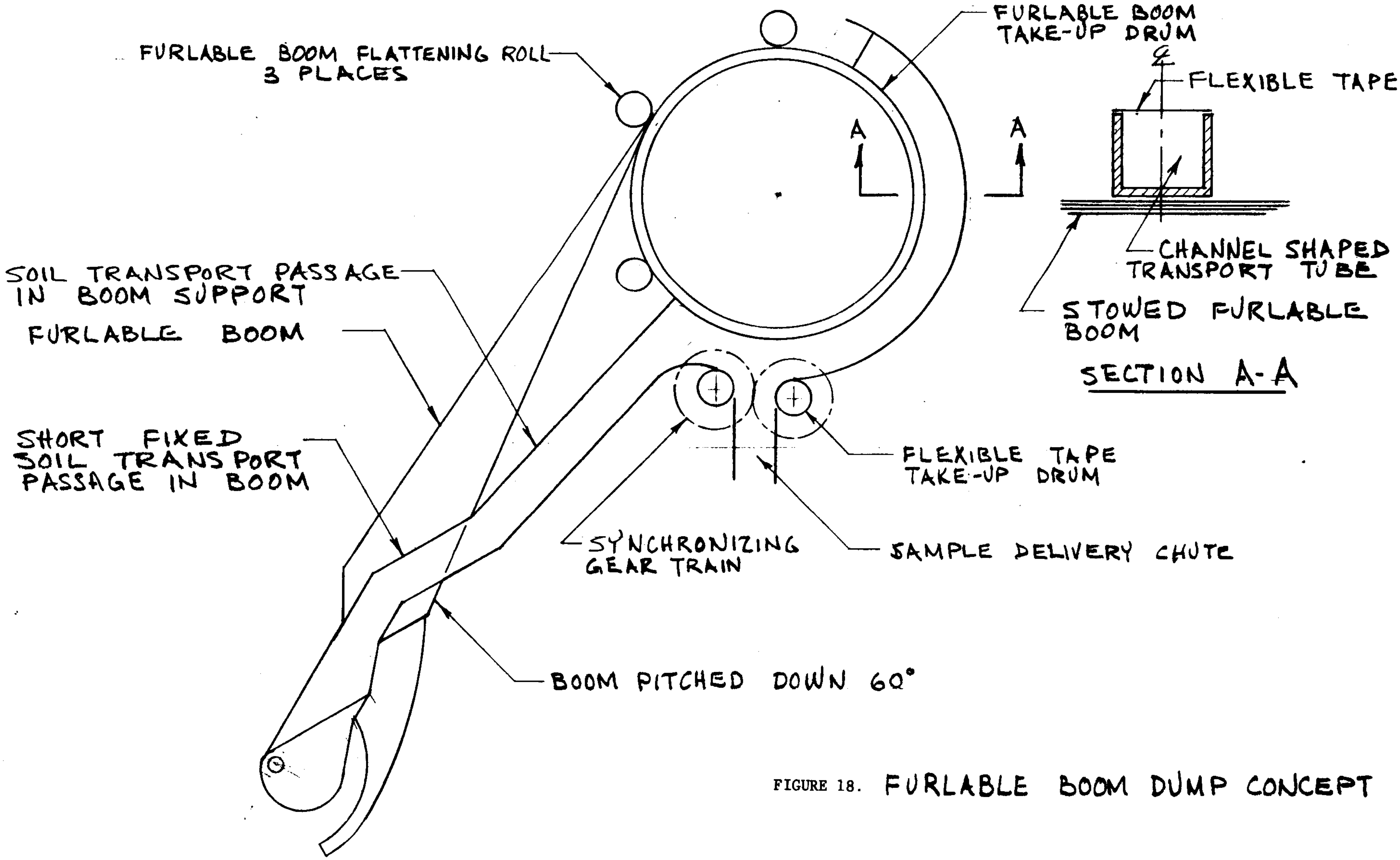


FIGURE 18. FURLABLE BOOM DUMP CONCEPT

deformation during stowage is small. Thus, a short fixed tube can be incorporated between the backhoe scoop and the opening in the side of the boom. This soil transport tube can be sealed at the outlet in the side of the boom tube thereby preventing soil particles from gaining entry into the interior of the boom. When the boom is fully retracted, this opening is aligned with a corresponding one located in the boom tube guide shoe. This opening is the end of a tube leading to a channel shaped passage fitted around the boom tube take-up drum. The fourth side of this passage is closed with two flexible tapes. The take-up spools for these tapes are mounted on either side of the sample delivery chute. These take-up spools are geared together so that as the boom is erected to a vertical position, tape is deployed from one and taken up on the other. This in effect provides an opening in the sample delivery passage that remains fixed with respect to the sample delivery chute regardless of the orientation of the boom. Any soil remaining in the channel shaped passage after the soil is dumped is progressively dumped into the sample delivery chute as the boom is lowered from the vertical. Wipers can be installed at the take-up spools to ensure that no particles remain adhering to the tape.

In attempting to mechanize this approach, primary attention was given to compactness, since the flattened boom is rather wide and required a fairly large diameter take-up drum. The demonstration model of this boom fabricated by Ryan Aeronautical Co. had the physical characteristics listed in Table III.

TABLE III

PHYSICAL CHARACTERISTICS OF RYAN
FURLABLE BOOM

Characteristic	Description
Boom material	6Al 4V (Titanium Alloy)
Flattened boom width	4.5 inches
Wall thickness of boom	.006 inches
Drum diameter	7.0 inches
Boom length	120 inches

This boom was built to the specification that it be capable of supporting a one pound weight at the tip when fully extended and that it be capable of applying a downward force of two pounds under the same conditions. Thus, the cable scoop actuation system was incorporated in this approach in the interest of minimizing the boom tip weight.

It appeared that the diameter of the take-up drum might conceivably be reduced, improving the volumetric efficiency of the design. The analysis for allowable flattening stresses, presented in Section 2.3.1 of Philco-Ford Report Number UG-3962, was used to check the required take-up drum diameter. The material properties for several materials are compared in Table IV.

TABLE IV

MATERIAL PROPERTIES COMPARISON

MATERIAL	PROPERTY	VALUE
6Al 4V Titanium Alloy	Yield point Young's Modulus E/σ	120,000 psi 16×10^6 psi 133
PH 15-7Mo Steel	Yield point Young's Modulus E/σ	145,000 psi 29×10^6 psi 200
Fiberglass	Yield point Young's Modulus E/σ	50,000 psi 5×10^6 psi 100
Beryllium/Copper	Yield point Young's Modulus E/σ	120,000 psi 19×10^6 psi 158

The lower the value of E/σ, the more deformation can be tolerated without yielding the material. It is seen that the titanium alloy falls between fiberglass and beryllium copper making it a good choice for the boom material when considering its relatively lower density when compared to Be/Cu or steel. An allowable ratio of tube diameter to wall thickness of 150 is reasonable for this material. Since this boom consists of two sheets welded at the edges, the effective wall thickness in terms of flexing is twice the thickness of the individual sheets. This results in a nominal tube diameter of $d = (.012)(150) = 1.8$ inches which is very near that used in the Ryan boom. Since this boom has curvatures of the tape in both directions, it must be considered in the category of a backward wound tape which is more severe in stressing the material. For this type of winding, the stress falls rapidly with increased drum diameter. A good drum to tube diameter has a ratio of 3:1. Larger drum diameters do not appreciably reduce the stowage stresses. This results in a drum diameter of $D = (1.8)(3) = 5.4$ inches. Since the analysis used is conservative, a drum diameter of 5 inches was assumed for the initial design.

Integrating the cable actuation system with the drum design, it was apparent that it would be desirable to make the cable take-up drum nominally the same as the boom take-up drum. Since a small pulley or drive drum at the tip of the boom is desirable, a one-to-one ratio did not appear feasible. The geometry relating the drive drum response to the boom attitude for drums of different diameters is shown in Figure 19.

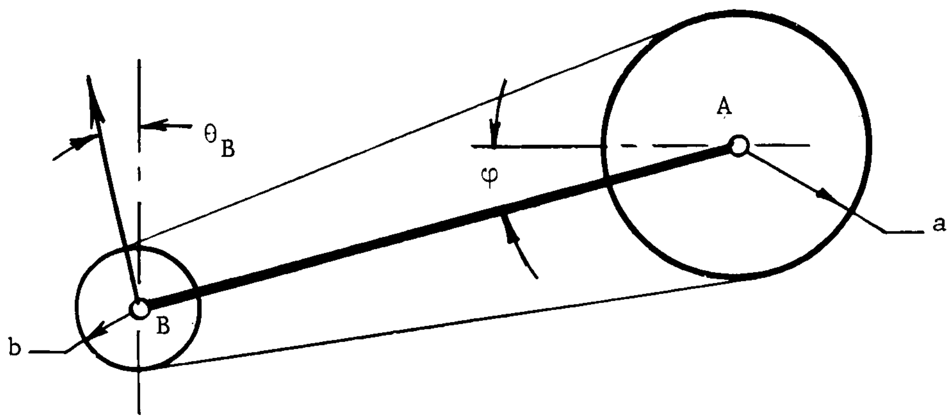


FIGURE 19. SCOOP ACTUATION CABLE GEOMETRY.

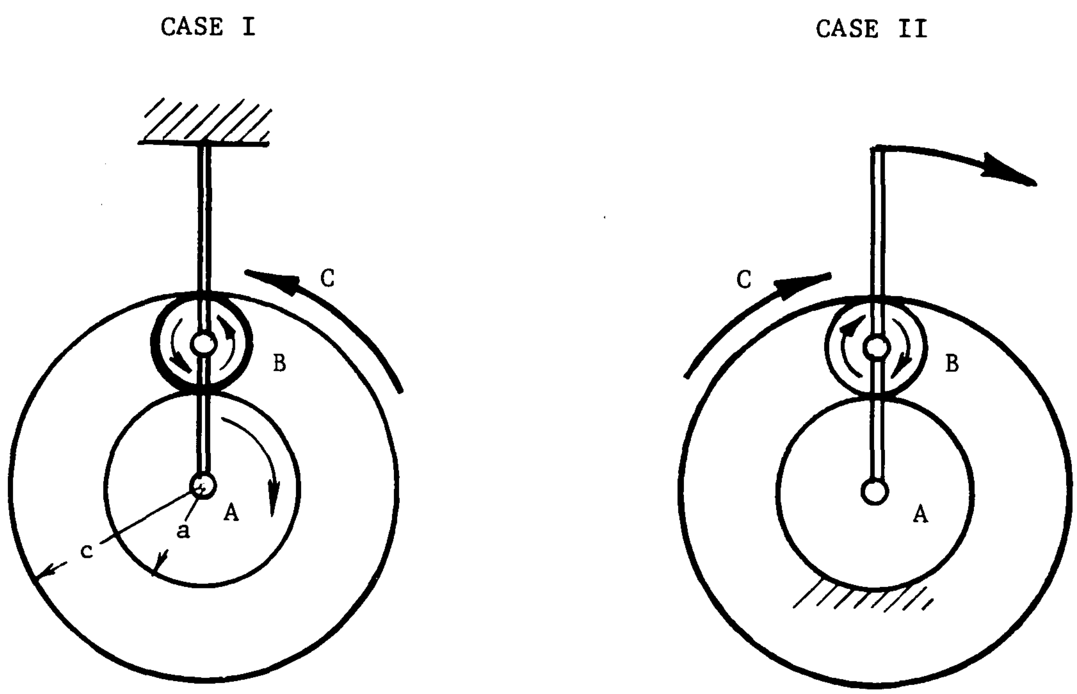
The rotation of B with respect to A is given by $\theta_B = \theta_A\ (a/b)$. If a is fixed and the arm AB moves, then $\theta_A = -\varphi$ and $\theta_B = \varphi - \varphi(a/b) = \varphi(1 - a/b)$. From the preceding equation, it can be seen that when a = b, the rotation of B is zero; i.e. it maintains a fixed rotational orientation with respect to A. If a larger drum diameter at A is used, then the rotation of B is given by

$$\theta_B = \varphi\ (1 - \frac{a}{b}) = -k\varphi.$$

That is, B rotates clockwise when φ is a counterclockwise rotation and the rotation of B differs from φ by a constant of proportionality. Thus, gearing at B can be used to restore the condition achieved when a = b by using a gear ratio to step up the response of B to equal that of A. If a diameter of two inches is assumed for B, then the required gear ratio at the scoop is

$$\frac{\theta A}{\varphi} = 1 - \frac{a}{b} = 1 - \frac{5}{2} = -1.5.$$

Since the drive drum at B is over-responding, a gear reduction must be incorporated as shown in Figure 16 between the scoop and the cable drive drum. The parallel linkage on cables must extend synchronously with the boom and simultaneously be capable of actuating the scoop. The extension can be accomplished using two drums rotating in opposite directions to feed out the upper and lower cable as required by the boom extension. Rotating the two drums together will cause B to rotate thereby actuating the scoop. A dual input drive to the drums are indicated and can be achieved with a planetary gear arrangement as shown in the sketch in Figure 20.



$$\omega_C = \omega_A \frac{a}{c}$$

$$\omega_C = \omega_D \left(1 + \frac{a}{c}\right)$$

FIGURE 20. PLANETARY GEAR SCHEMATIC.

Case I represents the condition existing when the boom is being extended or retracted to drive the cable take-up drum. Since two drums rotating in opposite directions are required, two identical gear trains as shown for Case I must be employed. The rotational input at A must have an opposite sense. Case II represents the situation when A is not driving but the planetary carrier arm is driving. In this case C rotates in the same sense as the carrier arm D. If this planetary carrier arm is common to both gear trains, then both drums will rotate together. For the assumed geometry the rotation of C is given by

$$\omega_C = \omega_D \left(1 + \frac{2}{1}\right) = 3\omega_D.$$

Since the angle that the scoop must go through to open is 60 degrees, the planetary carrier arm is required to move through an angle of 20 degrees. The conditions pertaining for Case I and Case II can be superimposed to obtain the case where scoop actuation occurs simultaneously with boom deployment. In either case, the planetary carrier arm must move only between two fixed points with respect to the boom support structure, thus allowing the use of microswitches to sense whether the scoop is open or closed.

Figure 21 shows a partial section through the take-up drum assembly for this boom. The boom extension/retract drive motor is mounted inside the take-up drum on a hollow shaft tied to the boom housing structure. This motor simultaneously drives the boom take-up drum through a ring gear attached to the inside of the drum and a geared shaft leading to the planetary drive of the inner cable take-up drum which rotates in the same direction as the boom take-up drum and at the same rate. The ring gear inside the boom take-up drum drives the input gear to the outer cable take-up drum through a set of planetary idlers mounted on the drum housing structure. The idlers effect a reversal of input rotation causing the outer gear to drive in the opposite sense as the inner drum but at the same rate. A ring gear affixed to this outer cable take-up drum also drives the boom flattening rolls. The idler rolls should be driven so that they keep the furlable boom tightly wrapped on the take-up drum. This can be accomplished by driving them so they try to feed slightly faster than the take-up drum. This implies that they must be elastically mounted to the shaft to allow for the differential motion if no slippage occurs. If the gear ratio is set so that the lineal surface speed of the idler roll and take-up drum match at the start of deployment, an increasing feed rate for the idler roll is achieved due to the fact that the take-up drum diameter decreases as the boom is deployed because the numer of flattened wraps on the drum is less. The idler rolls are driven through a gear train and are hinged at the axis of the idler gear between the driving ring gear and the gear on the flattening roll. This allows the flattening roll to pivot as required to accommodate the change in drum diameter as the boom is deployed. The flattening rolls are held against the boom take-up drum

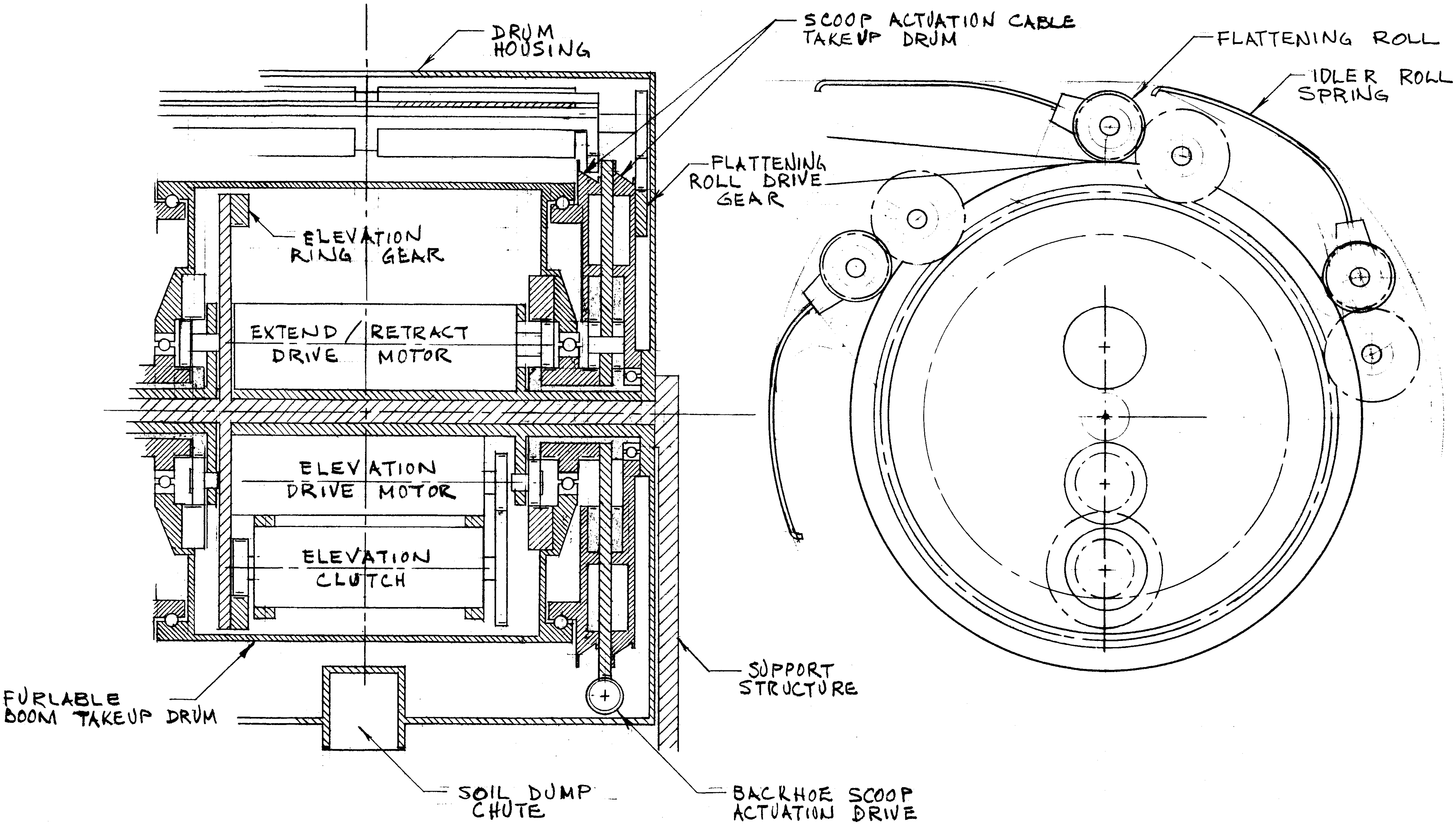


FIGURE 21. FURLABLE BOOM DRIVE CONCEPT

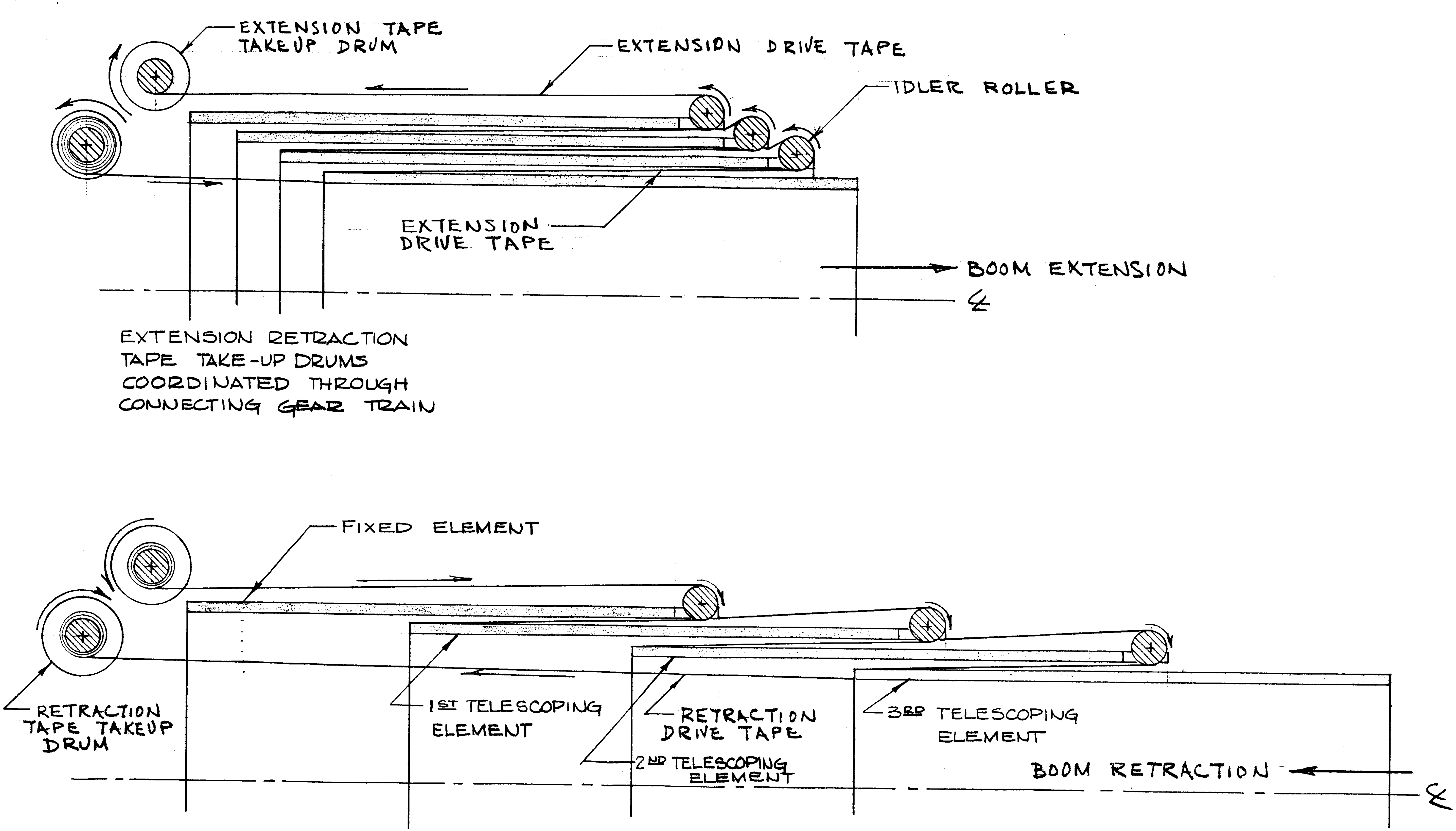
by means of flat wide cantilever springs attached to the roll support structure and bearing against the housing structure.

The elevation drive motor is also mounted inside the boom take-up drum on the drum housing structure. This motor drives through a ring gear attached to the fixed support structure about which the boom housing is free to pivot. Thus, the gear output at the elevation clutch walks around the ring gear causing the boom housing structure to be carried with it to control the elevation. The elevation clutch is incorporated so that a sampling traverse can be made with only the weight of the boom acting on the scoop. Alternatively, a load sensor can be incorporated to sense the vertical force being exerted on the scoop to control the rate at which the elevation drive motor operates to maintain a limited downward preload on the scoop.

This mechanism results in a drum housing of 7 inches in diameter and 6 inches wide. The transition length of the boom is approximately one foot so that the overall length from the center of the drum to the backshoe scoop is approximately 14 inches.

3.5.3 MECHANICALLY EXTENDIBLE TELESCOPING BOOM

Another approach considered the use of a telescoping boom using a completely mechanical means of extending and retracting the boom. Such a boom is shown schematically in Figure 22. In this concept, an idler roller is mounted on the tip of all but the last tube segment. A thin tape or small diameter flexible cable is attached to the preceding tube segment, passed over the idler roller back between the tube segments, and is attached to the base of the next tube segment. This is repeated with each tube segment until all the telescoping elements are interconnected. The tape from the first telescoping element is carried back and attached to a tape take-up drum. When power is applied to the take-up drum to rotate it, the tape pulls on the base of the first telescoping segment causing it to extend relative to the fixed element. This in turn causes a tensile force to be applied to the tape connected to the base of the second telescoping segment causing it to also extend. The same occurs for the third telescoping segment. Thus, all segments are being extended simultaneously at the same rate relative to the adjacent segment. A closed cable system can be made by attaching another tape or cable to the base of the third or final telescoping segment and attaching the other end to another take-up drum for retraction. By appropriately sizing the drums and interconnecting the extension take-up with the retraction take-up through a gear train, the deployment and take-up of the tapes can be coordinated so that the tapes or cables will always be taut during the extension or retraction of the boom. A boom of this type offers the advantages of higher strength and rigidity and will also allow a gravity dump to be made at any point in the extension cycle. It also allows the use of various cross-sectional shapes for the telescoping elements, such as square or triangular, if desired. A preliminary configuration of this boom as applied to the backhoe sampler is shown in Figure 23.



FOLDOUT FRAME 1

FIGURE 22. MECHANICALLY EXTENDIBLE BOOM

FOLDOUT FRAME 2

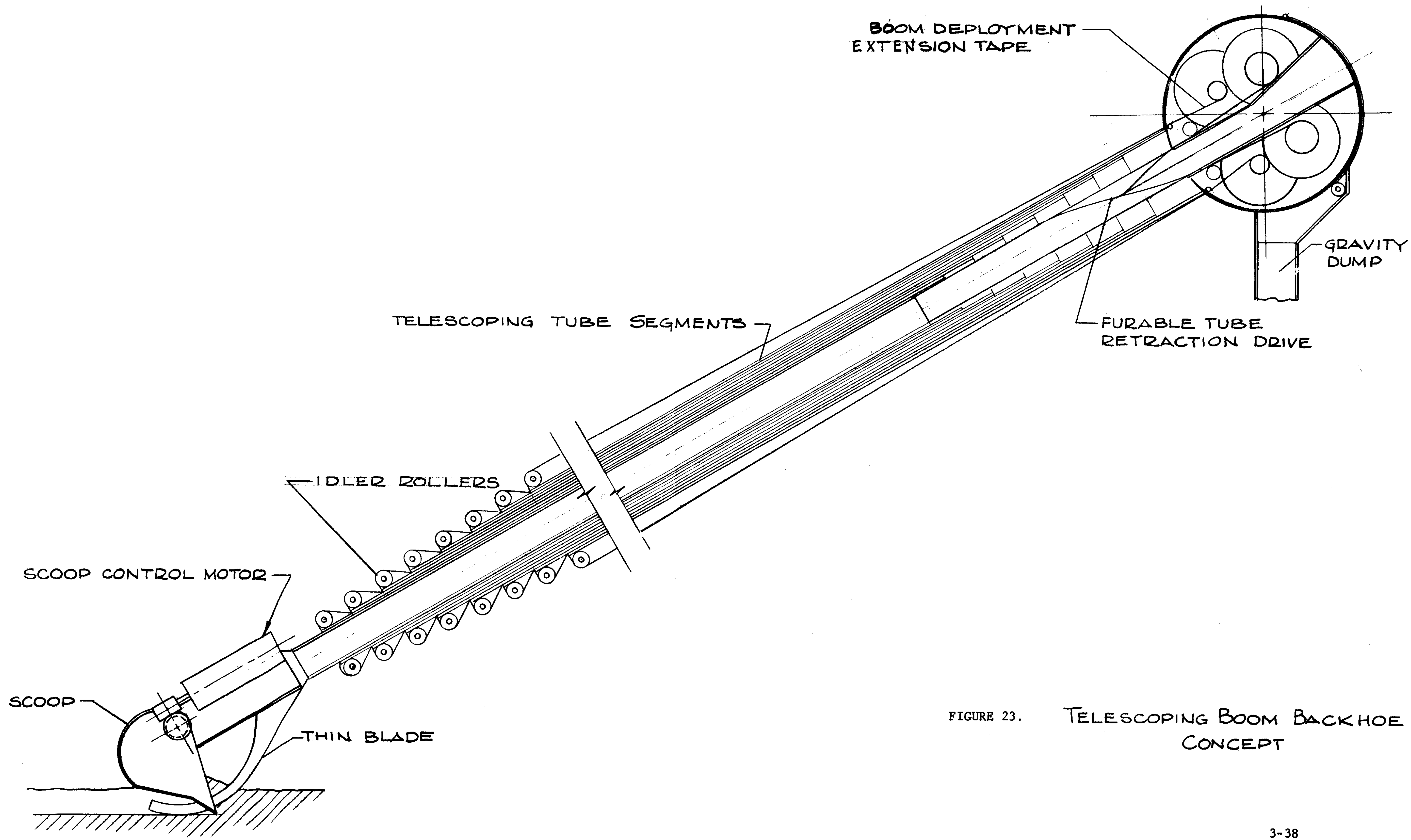


FIGURE 23. TELESCOPING BOOM BACKHOE CONCEPT

The retraction tapes are furlable tubes utilizing the principle of the DeHavilland furlable tubes. As the boom extends, these two tapes form a secondary tube inside the telescoping boom structure. This boom offers the following advantages:

(a) The internal tube can be used to transport soil at any extended position of the boom simply by raising the boom to a vertical position.

(b) The telescoping boom is inherently stronger than a furlable boom since there are no restrictions governing wall thickness, as there is for the furlable tube.

(c) The secondary internal tube protects the sliding segments of the boom internally from soil particles during the dump mode.

(d) Depending on the total extension required, this boom can be more easily configured to occupy a small volume when stowed.

It should be pointed out that the cable scoop actuation system can be used with this boom also; however, the greater simplicity associated with the motor drive would probably dictate its use even with the penalty of an increased reaction torque in the elevation mechanism. The increase in bending moment in the boom would not be critical from a structural viewpoint. The use of flat tapes rather than cables will allow a more compact design to be made in that smaller clearances can be used between telescoping segments and smaller diameter idler rolls can be used. The extension tape take-up drums are coordinated through a geared drive with the retraction tape take-up drums so that these tapes are deployed simultaneously at essentially the same rate. It may be necessary to mount the take-up drums on their shafts with springs to provide a preload to take up small differences in extension rates since the extension tape is much shorter than the retraction tape resulting in differences in the effective take-up drum diameters. This configuration is felt to be much simpler to mechanize and should occupy a small stowed volume.

3.5.6 SOIL AUGER SAMPLER, E-6

This sampler mechanism consists of an auger 1.25 inches in diameter and about 4.5 inches long. The auger is rotated slowly while thrust is applied to cause it to penetrate the soil. After penetration is achieved, the soil auger rotation is stopped and the auger is withdrawn without rotation to its initial stowed position. When it reaches this point the auger is spun at a high rotational speed to spin the soil off of the auger flights. Two basic approaches to mechanizing this sampler were taken as discussed in the following.

A variation of the canted feed roller system, developed by Philco-Ford under a previous contract, was applied to this sampler as shown in Figure 24. In this approach, one drive motor with a coaxial output from the gearbox is used to simultaneously drive a high speed gear train for the spin dump and a low speed gear train to drive the axial feed and rotate the auger as required. The low speed gear is connected to a hollow shaft with flats on the outer surface as shown in section A-A of Figure 24. The feed assembly housing has a hole shaped to fit this shaft so that the low speed gear train is always driving the feed housing regardless of direction of rotation or axial position along the shaft. When the direction of rotation is such that downward axial feed is achieved, the over-running clutch engages the shaft of the auger cuasing it to rotate with the housing. The feed is achieved by six rollers mounted to the feed assembly housing in sets of three rollers at the upper and lower end of the housing. These rollers are canted at a small angle and are mounted on torsion bars in such a manner that they are pressed tightly against the outer support tube. Thus, rotation of this assembly causes the canted rollers to feed the auger up or down, depending on the direction of rotation. These rollers are mounted on the ends of torsion bar supports so that, as the axial thrust builds up, the reaction forces acting on the feed rollers causes them to rotate to a smaller cant angle thereby reducing the feed rate. Thus, if a strong cohesive surface is encountered the axial thrust is built up to some maximum value determined by the torsion bar deflection characteristics. As the auger shears the soil the load is reduced and the canted feed rollers immediately move to an angle which will again cause the auger to feed into the soil. In this manner the feed rate always accommodates itself to a rate at which the auger can penetrate while maintaining the requisite axial thrust to cause penetration. To withdraw the auger the polarity of the power supplied to the motor is reversed thereby reversing the rotation of all gear trains. When this happens, the over-running clutch releases the auger shaft so that it is not driven while vertical feed is applied. When the auger mechanism reaches the initial stowed position, the conical tip of the high speed drive shaft is driven into a mating socket thereby engaging the high speed drive to spin the soil off of the auger and into the annular chamber at the base of the outer support tube. Some features of this design approach are as follows:

(a) The mechanization is simple and requires only one drive motor.

(b) Except for reversing the polarity of the power to the drive motor, the operational steps are automatically executed.

(c) Because the direction of rotation is dictated by the over-running clutch, initiation of the high speed spin will tend to cause the sample to feed down along the auger flights; however, the rapid increase of centrifugal force should spin the soil off before much downward feed takes place. Embossing the surface with shallow radial saw tooth slots would inhibit the downward feed.

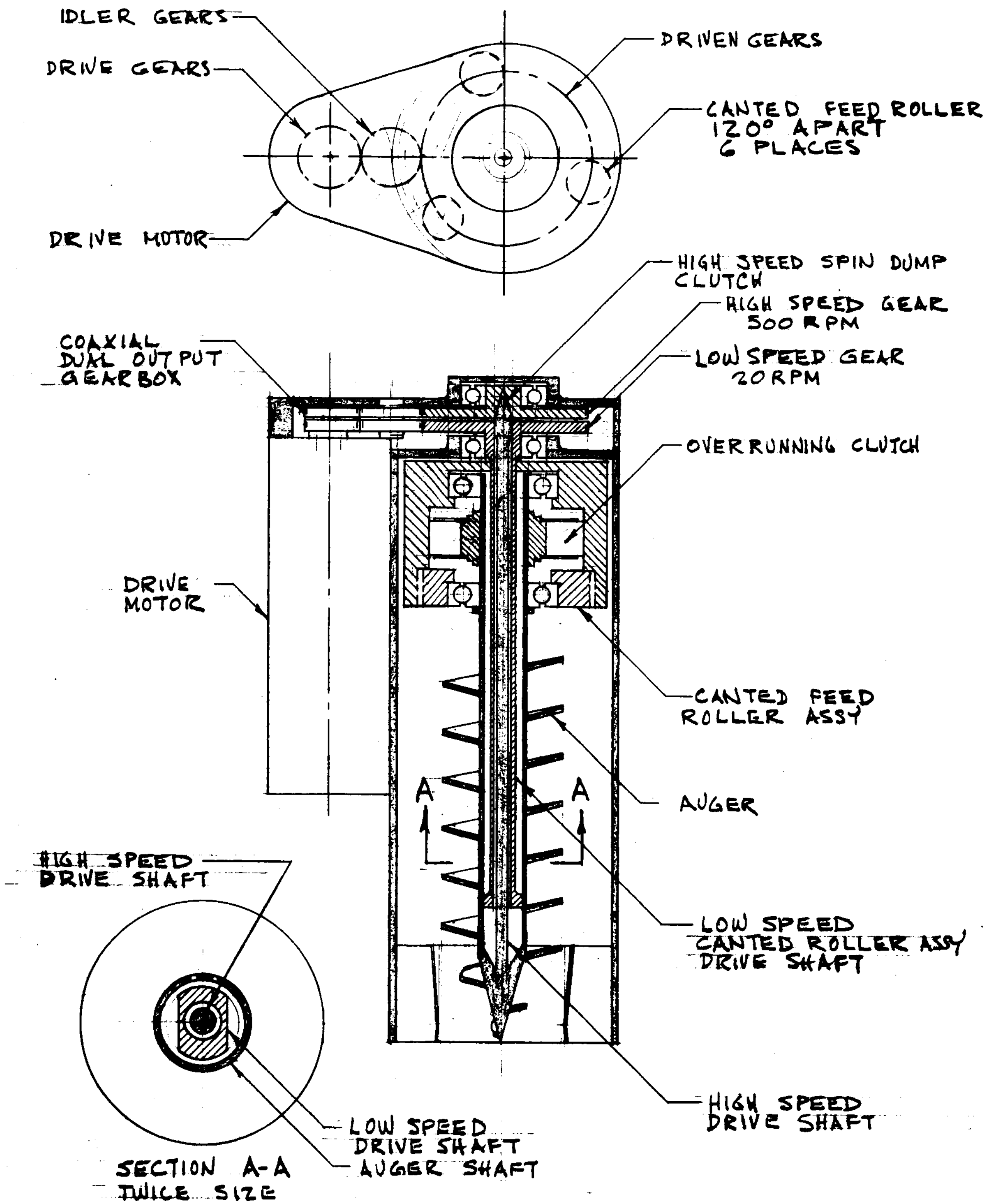


FIGURE 24. SOIL AUGER-CANTED FEED ROLL DRIVE

The second design approach uses a system more closely related to the breadboard version built by JPL as shown in Figure 25. Two motors are required, one to drive the axial feed screws and another with a coaxial output gearbox to drive the auger at low speed for digging and at high speed for the spin dump. The thrust exerted by the axial feed screws are reacted through springs at the top of the screws. As the thrust builds up the screws are lifted in proportion to the thrust. At some predetermined thrust limit, the ends of the screws actuate one or both of the snap action switches connected in series to turn off the power to the feed drive motor. As the thrust falls off the springs push the screws down until the switches are again closed, thereby providing power to the feed drive motor. Thus, axial thrust is maintained while the feed rate is adjusted to accommodate itself to the feed rate of the auger. The auger is engaged with the appropriate speed drive gear train through two overrunning clutches mounted on the auger drive shaft. The relative rotation and clutch engagements are shown schematically in Figure 26. The drive shaft is square in cross-section as shown in section A-A of Figure 25. This shaft is fitted into a square hole in the auger so that rotational power is applied as the auger advances. Some features of this design approach are as follows:

(a) The shaft of the auger can be smaller in diameter than for the canted feed roller approach.

(b) The operational reliability of the over-running clutch to engage the high speed spin is probably better than the cone friction clutch.

(c) More programmed inputs are required to complete an operational cycle; however, more flexibility for altering the operational mode exists.

(d) The axial feed lead screws are exposed to the soil during the spin dump. Some sort of shielding may be required.

(e) The auger carrier guide rolls could be eliminated by using three feed screws driven synchronously.

As originally presented to JPL, this auger was to be mounted on a telescoping boom. At their suggestion a short rigid boom will be used. The orientation of the soil auger with respect to the local vertical can be achieved by either a parallel bar linkage or a closed cable system. The closed cable system will probably provide the lightest configuration. Final transfer of the soil from the annular chamber in the outer support housing will be achieved by erecting the boom to a vertical position. Since the soil auger orientation is always maintained in a vertical position, this will position the bottom of the support housing or annular

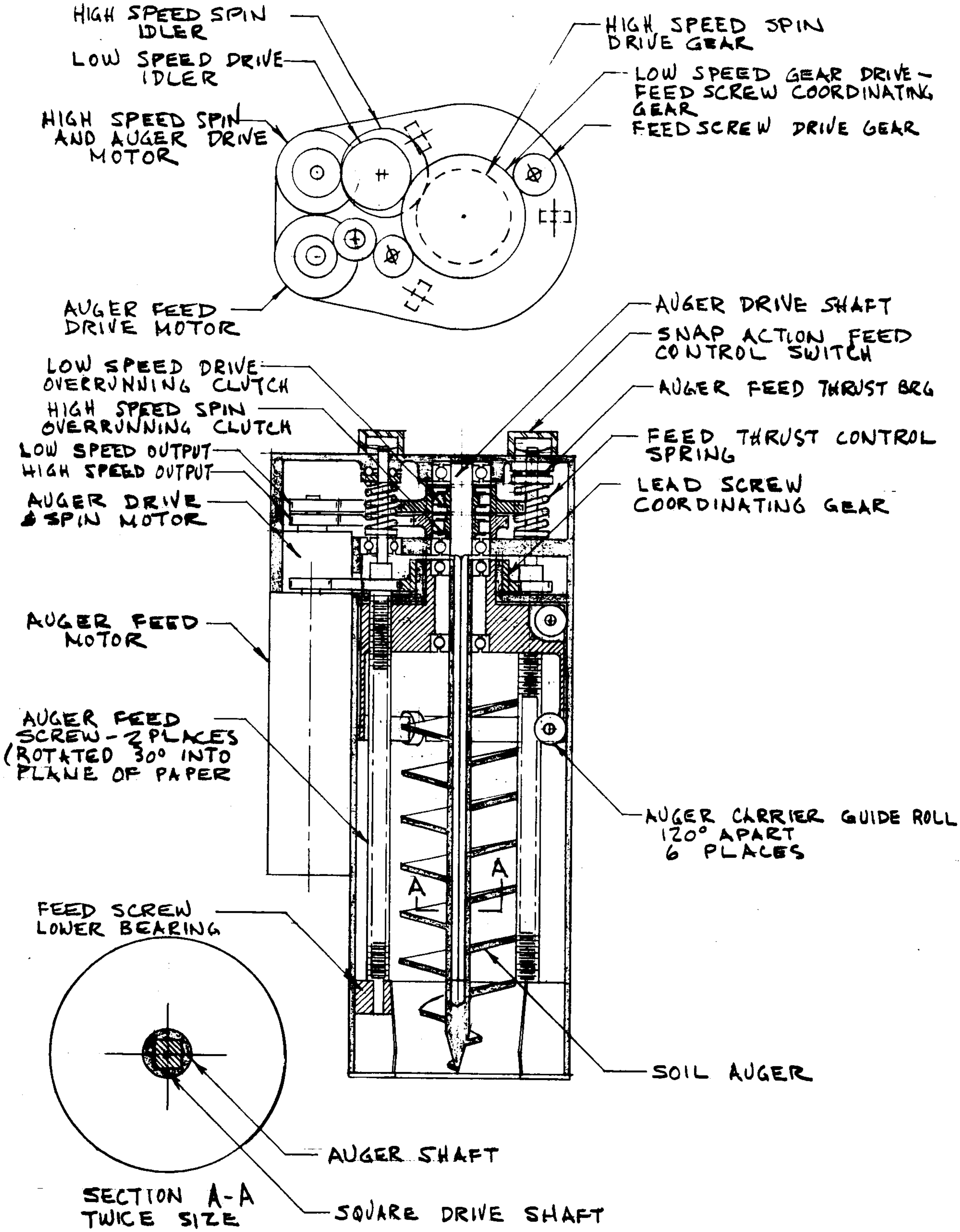
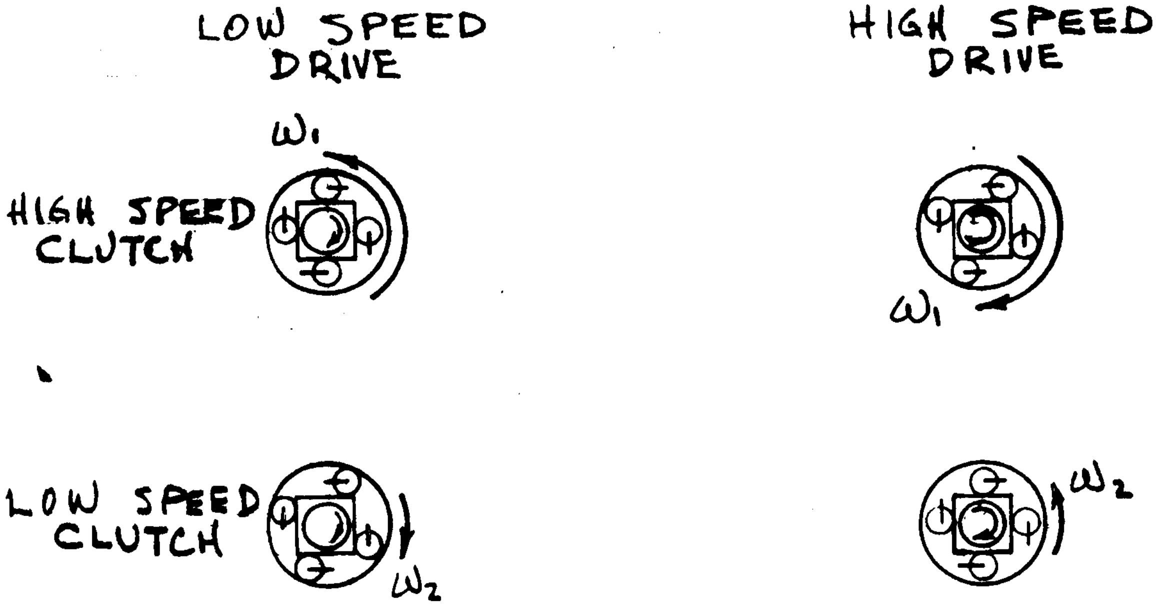


FIGURE 25. SOIL AUGER-LEAD SCREW FEED



ω_1 = 500 RPM ω_2 = 10 RPM

SCHEMATIC- CLUTCHING ARRANGEMENT
SOIL AUGER DRIVE - LEAD SCREW FEED

1. COAXIAL OUTPUT GEARBOX DRIVES BOTH HIGH SPEED AND LOW SPEED GEAR TRAIN
2. DURING LOW SPEED DRIVE OVERRUNNING CLUTCH ALLOWS HIGH SPEED GEAR TRAIN TO RUN IDLE
3. DRIVE MOTOR TURNED OFF DURING RETRACT CYCLE — CLUTCHES PREVENT AUGER ROTATION IN REVERSE DIRECTION DURING RETRACT
4. DURING SPIN DUMP OVER-RUNNING CLUTCH ENGAGES HIGH SPEED GEAR TRAIN AND RELEASES LOW SPEED GEAR TRAIN

FIGURE 26. SCHEMATIC OF CLUTCH ACTION

soil chamber over the end of the tubular boom. Two spring loaded doors closing the annular soil chamber are opened as the soil auger reaches the vertical position to dump the soil down the boom to the payload sample entry port.

3.5.7 MINIATURE JAW CRUSHER, E-7

The effort expended on this mechanism consisted of making an estimate of the forces required to crush a pebble with a maximum diameter of 5 millimeters. Two basic mechanisms can be used to develop a high mechanical advantage. These are a differential screw drive and a toggle mechanism. At first glance, the toggle mechanism appears more desirable since the friction forces in the differential screw threads could be high. Some typical properties of three rock types are given in Table V.

TABLE V

TYPICAL ROCK PROPERTIES

	Rock Type		
Property	Quartzite	Basalt	Granite
Poisson's ratio	.10	.25	.09-.20
Young's Modulus, psi	-	-	$5.8\text{-}8.7\times10^{6}$
Compressive strength, psi	$1.42\text{-}2.84\times10^{4}$	$2.84\text{-}4.98\times10^{4}$	$1.42\text{-}3.98\times10^{4}$
Tensile strength, psi	-	-	400-700
Shear strength, psi	-	-	2100-4300

Based on these values, a value for Young's modulus of 9×10^{6} psi and a compressive strength of 50,000 psi, was assumed in the calculations. A maximum force of 1600 pounds was calculated to be required on the jaw to crush a disc 5 millimeters in diameter. A design jaw force of 2000 pounds was used in all subsequent calculations. The jaw deflection required was calculated based on a sphere being compressed between two flat plates sufficiently to develop the maximum compressive stress across the cross section at the major diameter of the sphere. This should give a conservative value for the required deflection to cause fracture since the maximum stress at the point of contact is 460,000 psi. Timoshenko in Volume II "Strength of Materials" on page 357 quotes a typical value of 530,000 for a crucible steel ball. An order of magnitude less could be expected from rock based on the relative strength values for rock and steel. A required deflection or jaw movement of .04 inches was obtained from these calculations.

The jaw crusher mechanism envisioned in these calculations is shown schematically in Figure 3.

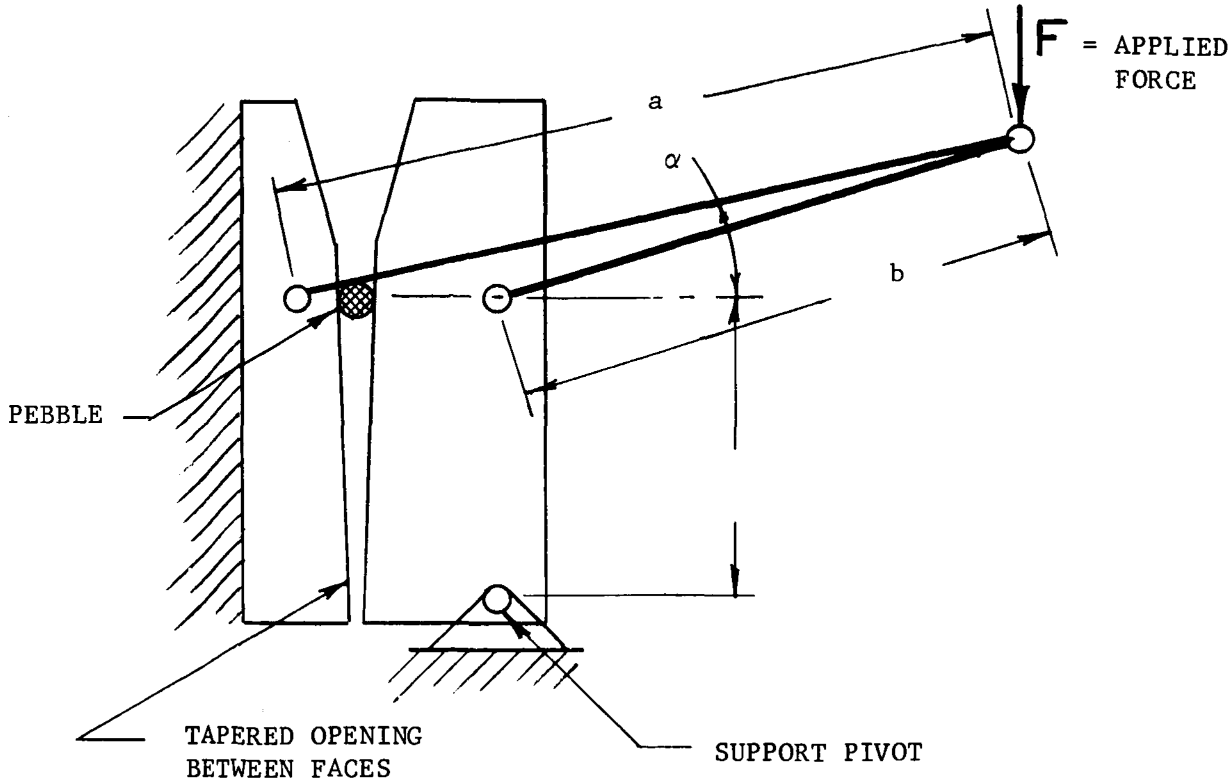


FIGURE 27. SCHEMATIC OF JAW CRUSHER MECHANISM

Equations were derived relating the required actuation force, the jaw force, and the geometry. Using these equations, the variation in required actuation force, F, and the required jaw movement, δ, were calculated for a set of values defining the geometry that might be typical of a miniature crusher.

These assumed values were a = 5 inches, b = 4 inches, c = 2 inches, and a jaw force of 2000 pounds. The variation of the required actuation force and jaw deflection with the toggle link angle α is given in Figure 27. It is seen that for the estimated properties of the pebble being crushed, an actuation force of 535 pounds and an initial toggle link angle of 14.4 degrees are required to produce a jaw movement of .04 inches with sufficient force to crush the pebble. It is also seen that reducing the required jaw

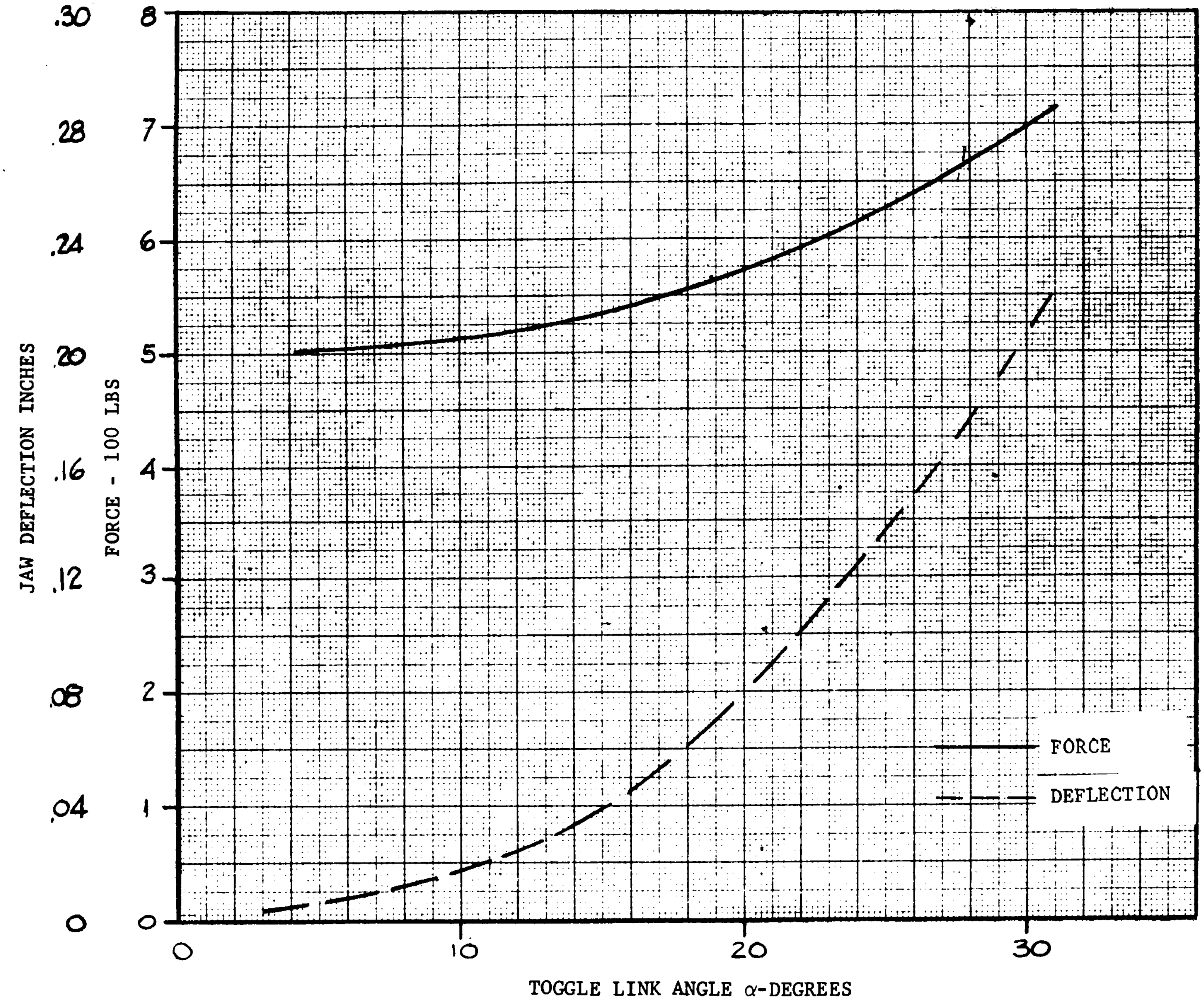


FIGURE 28. FORCE/DEFLECTION CHARACTERISTICS FOR JAW CRUSHER

deflection does not significantly reduce the required actuation force. The values obtained indicate such a mechanism is feasible; however, it should be pointed out that the assumed configuration does not represent an optimum. Other combinations of link/jaw geometry could produce more favorable relationships.

Since this initial work on the jaw crusher, the effort has been redirected to pursue a design for a miniature rotary crusher based on a breadboard model built and tested by JPL. The design criteria given in Section 2.0 for this mechanism reflect this redirection. Initially it was thought that the jaw crusher would be less susceptible to jamming by any malleable metallic material. This can be prevented in the rotary crusher, which is much more compact, by separating material with high magnetic permeability before it enters the crusher. Thus, a preliminary separation of material fed to the crusher will catch and hold the magnetic particles, which are presumed to be metallic meteroites. It will also separate the fine material below 300 microns in size and bypass these around the crusher to keep from loading the crusher and reduce the production of the very fine particle size population to a minimum.

3.5.8 PARTICLE SIZE SORTER, E-8

Several concepts for performing this processing operation were generated under the fundamental assumption that the size sorting is intended to prepare the sample for use in analytical instruments such as the petrographic microscope or the X-ray diffractometer. On this basis, the particle size sorter has to separate the raw sample into three cuts. These were defined in the design criteria for this mechanism to be $d > 500\mu$, $125\mu < d < 500\mu$, and $d < 125\mu$. Two basic approaches are available to perform a size sorting operation. One is to pass the material through a set of screens, as is conventionally done. The other is to utilize the effect of terminal velocity variation flux with particle size to separate particles using a pneumatic flow system. A disadvantage of the latter system is that size sorting is not accomplished independently of the material density. In addition to these two approaches either batch processing or continuous flow processing can be used. On this basis, four design concepts were generated and are shown schematically in Figure 28.

Concept (1) is a recirculating closed flow pneumatic system. It was assumed that for the low atmospheric pressures existing on Mars that this would have to be a sealed and pressurized system to operate. In operation the raw soil sample is metered out of the hopper in a dispersed condition and falls into the central tube. The upward flow velocity in this tube is sufficiently large to entrain all particles 500 microns in diameter or less. These are carried through the blower into the annular outer return chamber. This chamber is sized to reduce the flow velocity to a very low value allowing the particles to settle to the bottom of the chamber. Only

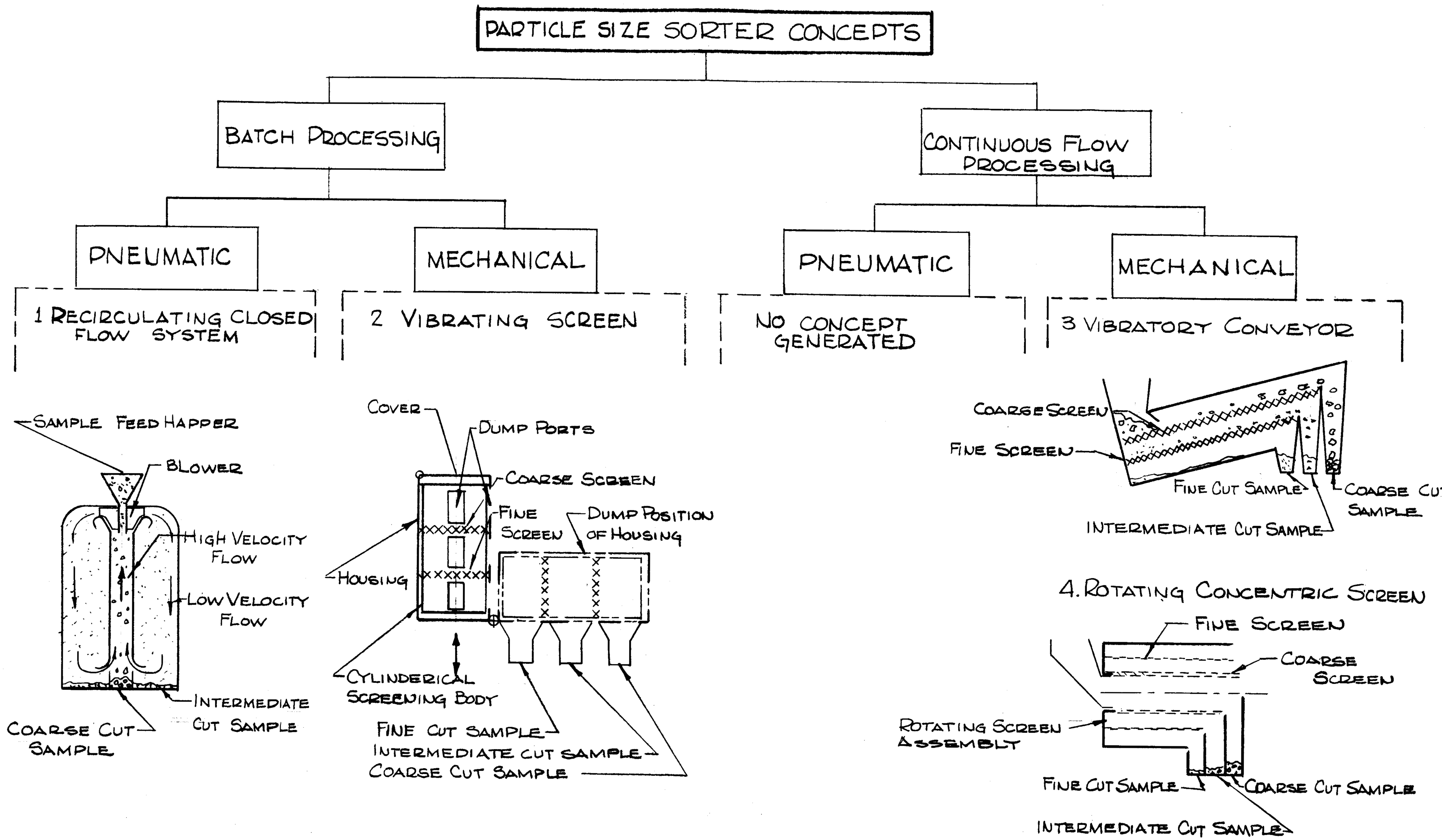


FIGURE 29. PRELIMINARY CONCEPTS

FOLDOUT FRAME 1

FOLDOUT FRAME 2

the very small particles obeying Stoke's Law remain entrained in the flow and continue to recirculate as long as the processing continues. When all the raw sample has been processed, the coarse cut and intermediate cuts are removed or dumped. A brush traversing the bottom of the annular chamber sweeps the intermediate cut to the dump port. After removing these two cuts, the exit ports are closed and the fine material is allowed to settle out, the majority of which will be on the floor of the annular chamber because of the larger volume involved. This fine cut can then be swept to another dump port and collected. It should be noted that the pressurized atmosphere of the system should be maintained during the dump or collection of the various cuts of sample.

Concept (2) is a conventional sieving system vibrated in a vertical direction. The sieves are mounted in a cylindrical body which has a close sliding fit inside a housing. The raw sample is introduced into the top chamber and the lid on the housing is closed. Vertical oscillation is then initiated and continued until sieving is complete. The housing is then pivoted to a horizontal position to transfer or dump the various cuts in each of the sieving chambers. The oscillating cylindrical body has ports cut into the wall which are blocked or closed by the close fitting housing during the sieving operation. To effect the sample dump, the cylindrical body is rotated inside the horizontal housing until the ports align themselves with corresponding ports in the housing. Oscillation of the cylindrical body continues during this operation causing the various cuts of sample to be shaken into the receiving hoppers or sample containers. The assembly is then returned to the vertical position for the next sieving cycle.

Concept (3) utilizes the concept of a vibrating conveyor to simultaneously provide the agitation required for sieving and the energy necessary to move the sample along the screen. The basic mechanism consists of a coarse screen mounted over a fine screen inside a rectangular container which is vibrated both vertically and horizontally in harmonic motion. The raw sample is metered onto the coarse screen which is then conveyed toward the higher end. As the sample is transported along the screen the finer material falls through. The completeness of sieving is determined by the dwell time on the screen. The various cuts are ultimately transported off the higher end of the screens or lower surface into their respective sample containers.

Concept (4) utilizes a series of concentric screens and an outer cylinder which is slowly rotated. The raw sample is metered into the inner cylindrical screen which is the coarse screen. The entire assembly is sloped at some slight downward angle so that as the sample is tumbled inside the screens, it not only falls through the screen but progresses along it toward the dump end where it falls into the appropriate containers. A wiper blade running inside the outer cylinder would prevent the fine sample from tumbling and aid in causing it to transport to the delivery end. An advantage of this system is that the surface area of the fine screen is much larger than the coarse screen thereby making more mesh openings available for the fine material. This should improve the sieving rate of the finer material which is known to become more difficult as the mesh size decreases. A disadvantage of this concept is that it is probably sensitive to orientation which might require a gimbal system to provide final orientation with respect to the local vertical.

These initial concepts were generated early in the task. After a review by JPL, it was indicated that this mechanism should be based on the existing breadboard particle size sorter used with the petrographic microscope being developed at JPL. The basic design approach used for this mechanism was to retain the fundamental features of the JPL breadboard design using screens canted at a 45 degree angle. Oscillation of the sieving chamber about a fixed axis produces not only vertical motion but also provides a centrifugal acceleration which will cause the particles to drift towards the sieving screen. The prime intent of this mechanization was to mechanize it in a more compact form and to expand its capability to serve other instruments as well as the petrographic microscope. To accomplish the latter goal, additional doors were incorporated into the bottom of each of the sieving chambers as well as those existing in the top which serve the petrographic microscope. This device is controlled through a mechanical programmer using cam actuation. This program can be subdivided into two subprograms, one which controls a sieving and dispensing cycle for the petrographic microscope and the other cycle controls a general purpose sieving cycle to serve some other instrument. The sequence of events for these cycles are given in Table VI. The proper cam program is engaged by means of over-running clutches depending on the direction of rotation of the drive motor.

After a review by JPL, it was indicated that the need to service the petrographic microscope should be deleted making this only a general purpose particle size sorter. Thus, the A part of the sequencing program can be deleted and the mechanism considerably simplified by the elimination of processing steps. Two approaches were used for the soil feed mechanism to load the sample weighing cup. One method used a vibratory conveyor to feed the sample to the cup. The other used an auger to feed the sample to the cup. JPL indicated that the vibratory feed mechanism would probably be too sensitive to orientation and that the auger feed appeared to be preferable. The auger feed will be used in the final design of this mechanism.

TABLE VI

PARTICLE SIZE SORTER SEQUENCE OF EVENTS

A. PETROGRAPHIC MICROSCOPE SIEVING CYCLE

1. Raw sample hopper is loaded from sampling mechanism, by external comand.
2. External command actuates soil feed mechanism to transfer sample to the weighing cup.
3. Weight sensor provides input to terminate soil feed when 1 gram is in cup and actuates main drive motor to start cam program and sieve oscillation.
4. Cam opens weighing cup dump doors dropping sample into sieve mechanism.
5. Cam closes dump doors (20 seconds).
6. Sieving cycle continues for 2 minutes.
7. Cam opens upper fine cut dump port and holds it open for 10 seconds to effect sample transfer to slide.
8. Cam closes upper fine cut dump port (20 seconds).
9. Cam opens upper intermediate cut dump port and holds it open for 10 seconds to effect sample transfer to slide.
10. Cam closes upper intermediate cut dump port (20 seconds).
11. Cam actuates switch to reverse polarity to main drive motor. (This releases petrographic microscope sieving program cams and engages general sieving program cams.)
12. Cam actuates dump port to transfer residue out of sieving mechanism.
13. Cam closes dump port and terminates operation.

B. GENERAL PURPOSE SIEVING CYCLE

1. External command actuates soil feed mechanism to transfer sample to weighing cup.
2. Weight sensor provides input to terminate soil feed when 5 grams are in cup and actuates main drive motor to start cam program and sieve oscillation.
3. Cam opens weighing cup dump doors dropping sample into sieve mechanism.
4. Cam closes dump doors (20 seconds).
5. Sieving cycle continues for 2 minutes.
6. Cam opens lower fine cut dump port and holds it open 10 seconds to effect sample transfer to receiving container.
7. Cam closes lower fine cut dump port (20 seconds).
8. Cam opens lower intermediate cut dump port and holds it open for 10 seconds to effect sample transfer to receiving container.
9. Cam closes lower intermediate cut dump port (20 seconds).
10. Cam actuates dump port to transfer residue out of sieving mechanism.
11. Cam closes dump port and terminates operation.

SECTION 4

PROGRAM STATUS

As of the last review meeting held at Philco-Ford, all design effort was reviewed and suggestions for the final design effort were made. These are itemized as follows:

4.1 UNCASED ROTARY/ IMPACT DRILL SAMPLER, E-1

(a) Do not use a mixture of pneumatic transport with mechanical transport of the sample. A mechanical transfer of the sample when the drill is in its stowed position will be used.

(b) The parallel bar linkage for deploying the drill was acceptable and will be used.

4.2 CASED ROTARY/IMPACT DRILL SAMPLER, E-2

No further effort on this design is to be pursued.

4.3 CONICAL ABRADING SIEVE CONE SAMPLER, E-3

(a) The intermittent pneumatic transport using the valved chamber in the acquisition head is preferred.

(b) Parallel bar deployment linkage would also be used with this sampler mechanism.

(c) The axial feed should be mounted on the same structure as the gimballed sampler so that thrust is always in the same direction as the hole being drilled.

4.4 HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER, E-4

Sampler, E-4

(a) The single bend helical conveyor configuration was acceptable for the final design.

(b) The size of the drive motor should be increased and mounted above and parallel to the helical conveyor housing.

(c) The tip cutter design should incorporate tungsten carbide inserts at the entrance to the conveyor housing to minimize wear. Also, a lower lead or pitch on the conveyor helix should be used at the entrance to the conveyor housing than for the rest of the conveyor.

4.5 BACKHOE SAMPLER, E-5

(a) Use the mechanically extendible boom with a maximum extension length of 5 feet.

(b) Consider the use of bands or tape drives rather than gear drives on the azimuth and elevation drives.

(c) Use the motor driven scoop with the motor mounted on the tip of the boom.

4.6 SOIL AUGER SAMPLER, E-6

(a) The lead screw feed approach was acceptable for the final design. Actually both approaches are applicable since no real difference in the configurational envelop exists.

(b) The deployment of this sampler will use a short rigid boom. Parallel cable system will be used to maintain the orientation of the sampler with respect to the local vertical.

(c) Consider the use of bands or tape drives on the elevation and azimuth drives.

4.7 MINIATURE ROTARY CRUSHER, E-7

(a) Use the dimensional configuration for the rotor of the JPL breadboard model in the final design.

(b) Incorporate a grizzly to separate large pebbles and provide a means of separating high permeability magnetic material.

(c) Bypass the fine material around the crusher rather than running it through the crusher.

4.8 SAMPLE SIZE SORTER, E-8

(a) Eliminate the petrographic microscope function. This will be only a general purpose sorting mechanism.

(b) The oscillating crank arm drive should act at the center of percussion to minimize production of excessive vibration because of increased reactive forces at the pivot.

(c) There is no need to close the sample inlet port.

(d) Doors must be closed very tightly with a preload to prevent sample loss during the sieving cycle.

It is projected that all final designs will be completed by 8 May 1968.